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GUIDANCE AND ACTUATION TECHNIQUES FOR AN ADAPTIVELY CONTROLLED VEHICLE

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August 15, 1982

INTERIM TECHNICAL REPORT FOR PERIOD 1 FEBRUARY 1982 - 31 JULY 1982

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Prepared For

Defense Advanced Research Projects Agency
1400 Wilson Boulevard
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August 27, 1982

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Gentlemen:

Enclosed please find three (3) copies of our interim technical report on "Guidance and Actuation Techniques for an Adaptively Controlled Vehicle". This report covers the period 1 February - 31 July 1982 on contract MDA903-82-C-0149, DARPA order number 4385.

Sincerely,

Barry J. Brownstein
Manager
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Microcomputer Systems Group

BJB:ngp

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This technical report describes the progress for the first six months of a twelve month project to support the ~~overall DARPA~~ Adaptive-Suspension Vehicle program. The Battelle project consists of two major tasks. The first task relates to vehicle guidance; the objective is to derive a method to enable an active-suspension vehicle to traverse rough terrain. The second task is concerned with actuation techniques for the legs on the vehicle, specifically an in-depth analysis, design, and demonstration of the shank hydraulic circuit. This report describes progress on the development of guidance algorithms and the shank hydraulic circuit.

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SUMMARY

This technical report describes the progress for the first six months of a twelve month project to support the overall DARPA Adaptive-Suspension Vehicle program. The Battelle project consists of two major tasks. The first task relates to vehicle guidance; the objective is to derive a method to enable an active-suspension vehicle to traverse rough terrain. The second task is concerned with actuation techniques for the legs on the vehicle, specifically an in-depth analysis, design, and demonstration of the shank hydraulic circuit. While the two tasks are closely related, each is discussed separately in this report.

Guidance Task

The guidance problem requires the development of algorithms which will allow the vehicle to decide when to lift its legs and when and where to place them, based on information received from a terrain-scanning system. The guidance problem has been approached by developing a graphic simulation with which to test the algorithms derived. The simulation provides a realistic representation of both the terrain-scanning system and the vehicle itself, both of which are described in the main body of this report. In addition, the simulation display includes information which allows indicators (such as vehicle stability) of the algorithms' performance to be monitored.

A survey of previous work in the area of legged vehicle locomotion over rough terrain has shown that the earlier approaches to the gait generation problem have generally been designed and optimized for one particular kind of terrain. Foothold selection with those earlier methods has been accomplished by searching the areas which the legs can reach to determine the "best" foothold.

In the present work, the goal has been to derive algorithms which will work well not just on one kind of terrain, but on terrain which ranges from quite smooth to very rough. Ideally, the algorithms will produce "free gaits" (similar to those derived in previous work) on rough terrain,

but will produce optimally stable periodic gaits on smooth terrain. Also in contrast to previous work, the foothold selection processes have been designed to be driven by first determining where it would be desirable to place the feet and then by working to make those placements possible.

The implementation of these ideas which is currently being investigated involves as its first step the determination of those areas which are suitable for footholds along the vehicle's projected path. Then a sequence of leg liftings and placements which will allow stable locomotion along that path is determined. It can be seen that with this approach the leg sequencing is based directly on the terrain characteristics and projected vehicle position. It is believed that this approach will eliminate much of the searching involved in earlier methods, and will also provide better performance in most kinds of terrain.

Work on this task during the second half of the project will involve the further development and improvement of the stepping algorithm which has been developed, as well as the investigation of alternative algorithms.

Actuation Task

The crux of the actuation task is to design and demonstrate an hydraulic actuation system for each of the legs on the active-suspension vehicle. The design requirements for leg actuation were initially established through discussions between Battelle and Professor Kenneth Waldron of The Ohio State University. These requirements were based on the original O.S.U. leg design, and consist of requirements for actuating the legs during vehicle cruise, turning or crabbing maneuvers, and emergency operation.

Battelle's actuation task consists of four technical phases. In the first phase a preliminary shank circuit demonstration unit was developed based on a preliminary analysis of the original O.S.U. shank circuit design. Following this task, a modified shank circuit design, currently under development, will be completed. As part of this development, a model of the shank hydraulic circuit will be developed to assist in the design optimization of the system. Following this, a computer-controlled preliminary shank circuit demonstration will be performed, leading to the fourth task, which is to assist O.S.U. in a demonstration of legged operation.

To date, the preliminary shank circuit demonstration is underway, as is the detailed design of final shank circuit. Results of the preliminary design are presented in the body of this report.

GUIDANCE AND ACTUATION TECHNIQUES FOR AN ADAPTIVELY CONTROLLED VEHICLE

by

M.R. Patterson, J.J. Reidy, and B.J. Brownstein

INTRODUCTION

It has long been recognized that most man-made vehicles are greatly inferior to human beings and other terrestrial animals in off-road locomotion. The shortcomings of current vehicles are particularly noticeable in the area of mobility. On rough terrain, a vehicle with a passive suspension system must accommodate obstacles by gross body motions and must push its way through soft soil. A system with active suspension units such as legs, on the other hand, can "pick its way" through rough terrain by selecting the most suitable footholds and stepping over obstacles and soft spots. In addition, a legged system can compensate for terrain irregularities on which it must step by actively adjusting leg heights, thus providing a much smoother ride.

It is apparent that the mobility advantages of a legged system depend on its ability to select suitable footholds and avoid unsuitable ones, as well as to then physically actuate the selected actions. The ability to guide the vehicle requires that such a system have some knowledge of the terrain in front of it and have the capacity to decide when it should lift its legs and when and where it should put them down. The design of a system (which will be called the guidance system) which would allow a man-made vehicle to make those decisions is one of the tasks of this project; the first part of this report describes the progress made on that task during the frist half of the project.

The main text of this part of the report contains, first, descriptions of the vehicle and the terrain-sensing system on which this work is based and the interfaces between those systems and the guidance system. The rest of this portion of the report is divided into sections based on the different topics of the work conducted so far. Those topics are the development of a terrain-vehicle simulation with which to test the guidance system, the generation of specifications for a terrain-sensing system to provide the required information to the guidance system, the determination of an appropriate approach to storing that information, the development of the stepping algorithms themselves (including a survey of previous work in the area), and the preliminary design of a hardware system to implement the guidance algorithms.

Once a foothold has been selected, the foot must be physically shifted to that spot, smoothly and accurately. With a target of a five mile per hour cruise speed, the requirements on the leg actuation system; the drive, shank and adduction-abduction circuits, are demanding. The shank circuit must also be capable of compensating for mistakes in foot placement, e.g. the foot slipping off of a rock. The development of the shank circuit design is the second task of this project. This portion of the report discusses the establishment of the system design requirements, the preliminary analyses of an original system design, the results of experimental evaluations of some of the system components, and a brief trade-off of candidate pumps for the final system.

GUIDANCE RESEARCH

Vehicle and Terrain-Sensing System

The vehicle on which this work is based is approximately fifteen feet (4.6 meters) long and four feet (1.2 meters) wide. It has six three-degree-of-freedom legs which are attached at the top of the vehicle and whose lengths can be varied between five and nine feet (1.5 and 2.7 meters). Thus the vehicle's height can also be varied from five to nine feet (1.5 to 2.7 meters). The legs' other two degrees of freedom are angular motions in both the forward-backward and abduction-adduction directions. The vehicle is expected to be limited, at least on rough terrain, to translational velocities of no more than eight feet/second (2.4 meters/second) and rotational velocities of no more than thirty degrees/second. Accelerations are expected to be limited to no more than four feet/second/second (1.2 meters/second/second) and fifteen degrees/second/second.

The terrain-sensing system which is used in this work is a range-finding system which is fixed at the front top of the vehicle. It scans in elevation and in azimuth and measures a range value for each of those (discrete) points in its scan range.

Stepping Algorithm Interfaces

From the terrain-sensing system, the guidance system will receive the measured range for each of the points which is scanned by the terrain-sensing system. It will also receive scan synchronization information which will allow it to determine the elevation and azimuth angles for each of the scan points. From the vehicle control system, the guidance system will receive vehicle translational and rotational positions and velocities together with foot positions and contact information.

The guidance system will send to the vehicle control system commands for foot lifting and placing with the associated foot trajectories for transfer-phase legs.

Terrain-Vehicle Simulation

For the purpose of testing the guidance algorithms being developed in this project, a graphic simulation of the vehicle and terrain has been developed. The simulation displays the moving vehicle and that portion of the terrain of which it is aware at any given time. The velocity of the simulated vehicle is controlled by means of a joystick.

The simulation is intended to provide as realistic a test to the guidance system as possible. Toward that end, it provides terrain scan data to the system in the same way as the terrain-scanning system described above would provide the data to a hardware implementation of the guidance system. That is, the data are provided to the algorithms in the same order as the actual scanner would provide them, and the data are those range values that the actual scanner would return, corrupted by expected random noise values. The simulation also provides a realistic representation of the vehicle described above, to the extent that it incorporates the dimensions of the vehicle body and legs, the limits of travel of the leg joints (and thus the reachable volumes of the legs), and the expected limits on the vehicle's velocity and acceleration. All of the parameters which define the characteristics of the terrain-sensing system and the vehicle can be easily changed to accommodate different designs for those systems.

In addition to the display of the vehicle and the terrain, the legs' support pattern (the convex polygon formed by the points representing the supporting legs' horizontal positions) is also displayed with the center of pressure for the vehicle. That display allows the changing stability margin of the vehicle to be monitored. The numerical value of the stability margin is also displayed, as are the times until the different legs will reach their limits and the slopes of the terrain at the footholds of the supporting legs. All this information is useful for evaluating the performance of the stepping algorithms.

Terrain-Sensing System Specifications

Specifications for the terrain-sensing system based on the expected requirements of the guidance system have been generated. The guidance system requirements are based on the need for information for two purposes: the selection of footholds, and the accommodation of the vehicle body to large-scale terrain variations.

The first of these purposes, foothold selection, requires that all those areas where the vehicle may have to step be scanned at least once with the following requirements: a linear resolution of one-half the vehicle's foot size (that is, about four inches, or ten centimeters) for selection of safe footholds, the detection of objects of six inches, or fifteen centimeters, or greater height since that is the approximate upper limit of terrain accommodation for the vehicle's control system, and the sensing of terrain which has a slope of less than forty-five degrees since that is the maximum slope on which it is desirable to place the feet. The smooth (without excessive acceleration) accommodation of large-scale terrain obstacles requires that the vehicle obtain terrain data to approximately two body lengths (thirty feet, or nine meters) in front of the vehicle.

These requirements product the following approximate terrain-sensing system specifications: two scans per second, one degree angular resolution, scanning over a -15^{0} to -75^{0} elevation range, and scanning over a $\pm 30^{0}$ azimuth range ($\pm 45^{0}$ would allow turning with a smaller radius). An accuracy of one to two inches, or two to five centimeters, is probably sufficient.

Terrain Information Storage

The guidance system receives terrain data in the form of range measurements indexed by elevation and azimuth angles from the scanner, but the information is not convenient for the stepping algorithms to use in that form. To be of most value to the stepping algorithms, the terrain information should be in the form of elevation values indexed by their horizontal positions. To provide this form, the guidance system uses a terrain array divided into cells based on a horizontal plane; one scan point is stored per cell.

When a range value is received by the guidance system, it uses the elevation and azimuth angles of that range value together with the known position of the vehicle to convert the range value to an elevation value and store it in the terrain array. As the vehicle moves along, new points are used to replace old ones, so the guidance system "forgets" terrain when it has passed a certain distance beyond it.

Stepping Algorithms

Previous Work

For locomotion over relatively smooth ground, the problem of stepping algorithms is fairly well understood [1]; in most cases, a periodic gait (sequence for lifting and placing legs) and predetermined footholds can be used. Over rough terrain, however, the problem is more difficult. When the vehicle's path contains obstacles larger than those that can be handled by active terrain accommodation, the guidance system must use preview information about the terrain so it can make adjustments to the vehicle's movements to account for those objects.

As far as the authors of this report know, the first work which was done on automatic guidance of a legged vehicle over rough terrain is that of Okhotsimski and Platonov [2]. They consider obstacles which can be overcome by recalculating foothold locations without changing a periodic gait. For example, the problem of traversing a trench which is located where one of the vehicle's feet would normally be placed is solved by moving the foothold to one of the edges of the trench and adjusting the nearby footholds to provide a smooth transition.

It is apparent that when obstacles are large or numerous the vehicle's gait must be modified and the method of Ohkotsimski and Platonov cannot be used. The first work in this area was done by Kugushev and Jaroshevskij [3]. They introduced the "free gait", a gait in which leg motions are determined not from a periodic sequence but from knowledge of the current state of the vehicle, its kinematic limits, and the terrain over which it is walking. Central to their free gait algorithm is the concept of a foothold's existence segment, the portion of the vehicle's path of motion over which a particular leg can reach that foothold. At each iteration of the algorithm, the determination is made of which leg will reach the end of its existence segment soonest. That leg is then lifted if that can be done without making the vehicle unstable; if lifting the leg would make the vehicle unstable, another leg (or legs) is first moved to allow it to be lifted. Footholds for leg placement are apparently found by positing a vehicle location farther along its motion path and determining if a foothold can be found there which the leg can reach; if one cannot be found, the posited vehicle location is moved farther back toward the vehicle's actual location until a foothold can be found on which to place the leg.

Kugushev and Jaroshevskij's work was extended by McGhee and Iswandhi [4] at The Ohio State University. They introduced the concept of the kinematic margin of a foothold, which is the distance along the vehicle's path of motion from its present position to the point at which the leg at that foothold reaches its kinematic limit. They then use that concept to implement a free gait algorithm with which at each iteration the leg with the smallest kinematic margin is lifted. If necessary, another leg is placed in such a position as to allow the first leg to be lifted. Leg placement is accomplished by examining all available footholds for a particular leg and then placing the leg at the foothold with the greatest kinematic margin. It can be seen that McGhee and Iswandhi's algorithm lifts legs whenever it can and only places them when required for stability; the algorithm thus sacrifices stability for adaptability.

Both of the free gait algorithms discussed above assume a flat terrain with areas which cannot be used for footholds. Patterson [5] has considered the application of the free gait algorithm on a cylindrical surface, which presents several problems not present in two-dimensional terrain. The most difficult of those problems is that as the relative orientation and distance between the vehicle and the terrain change, the location, size and shape of the terrain area which a leg can reach changes. That area, which for most legs on flat terrain is a sector of an annulus, becomes very irregular in rough terrain; the solution used in this work is to determine a minimum reachable area based on the regular cylindrical terrain. Patterson also modified the kinematic margin concept to be based on the instantaneous velocity of the base of the leg instead of the predicted path of the vehicle; the kinematic margin is then easy to calculate even for vehicle motions such as turning in place. In addition, he modified the free gait algorithm in such a way that legs are only lifted when their kinematic margins fall below a threshold margin. Different settings of that threshold can then be used to optimize the algorithm for either stability or adaptability.

Present Work

In the previous work described above, the gait selection methods described were designed specifically for rough terrain. In the present work, the goal has been to derive a method which will work well on all kinds of terrain. This method will ideally be one which generates "free gaits" on rough terrain, but wave gaits (optimally stable periodic gaits) on smooth terrain. In the area of foothold selection, previous methods have chosen the "best" foothold by searching the legs' reachable areas. In the present work, the foothold selection process has been driven by first determining where it would be desirable to place the feet and then by working to make those placements possible.

The first attempt at implementing the methods of gait generation and foothold selection was that of sequencing legs based on the goal of moving the vehicle through a succession of those support patterns with the most inherent stability. With such an approach, appropriate selection of footholds for the legs comprising those support patterns could be used to provide stability over the particular path which the vehicle is following. It was decided not to use this approach because on terrain where many areas are unsuitable for footholds, it would be likely to require extensive searches both for acceptable sequences of support patterns and for footholds for the legs in those support patterns.

The approach which is currently being investigated involves as its first step the determination of those areas which are suitable for footholds along the vehicle's projected path. Then a sequence of leg liftings and placements which will allow stable locomotion along that path is determined. It can be seen that with this approach the concept of a support pattern is not explicitly used, and the leg sequencing is based directly on the terrain characteristics and projected vehicle position. It is believed that this approach will eliminate much of the searching involved in the first approach, and will also provide better performance in most kinds of terrain.

Preliminary Implementation Design

It has been determined that a hardware system to implement the guidance algorithms described above would require two processors because one processor's full capacity would be needed to perform the calculations required to transform to earth-fixed coordinates each of the many scan point range data transmitted by the scanning system. The other processor would then be used to perform the computations required for the gait generation and foothold selection processes.

ACTUATION RESEARCH

System Requirements

The design requirements for the shank circuit of the Adaptively Suspension Vehicle were initially established through discussions with Prof. Ken Waldron of O.S.U. These requirements were based on the original O.S.U. leg design and consisted of the following:

Cruise Mode

1) Rapid retraction - lift foot 2"-8" in 0.05-0.10 seconds (10-15% of return cycle)
2) Slow extension - allow foot to coast down or extend the foot under pressure, in approximately 0.30-0.40 seconds
3) Lock on - no extension of foot under load.

Turn or Crab

1) Small displacements at full load

Emergency

1) Rapid extension under no load

Additional system considerations included:

- Total cylinder stroke - 48 inches
- Stride frequency (cruise mode) - 1 hz
- Estimate weight of lower leg - 50 lbs

At Prof. Waldron's suggestion, a maximum load of 1800 lbs per foot was assumed. The original leg design upon which these requirements are based is shown in Figure 1. A schematic of the hydraulic circuit developed by O.S.U. for this original leg configuration is shown in Figure 2.

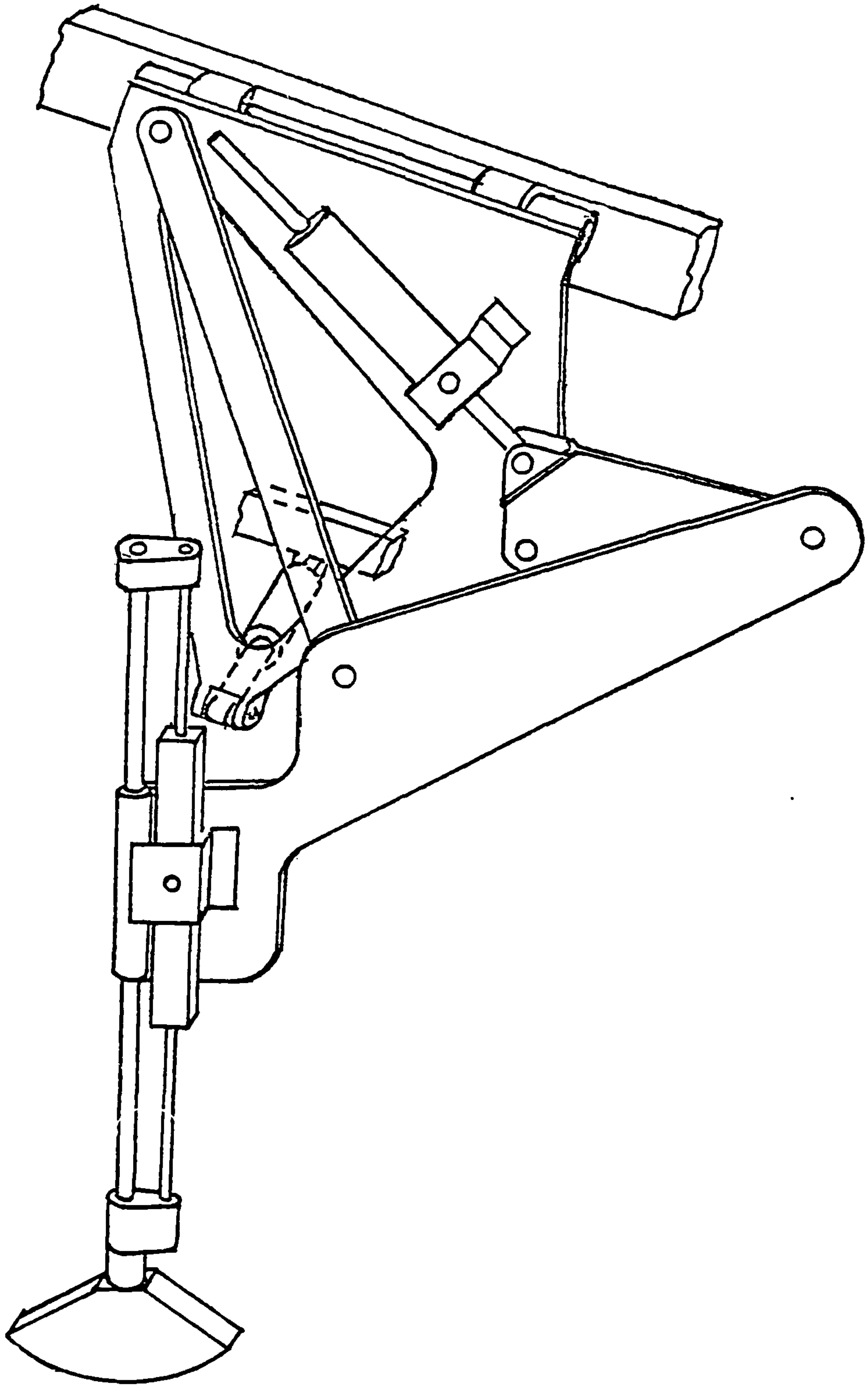

FIGURE 1. Original O.S.U. Leg Design

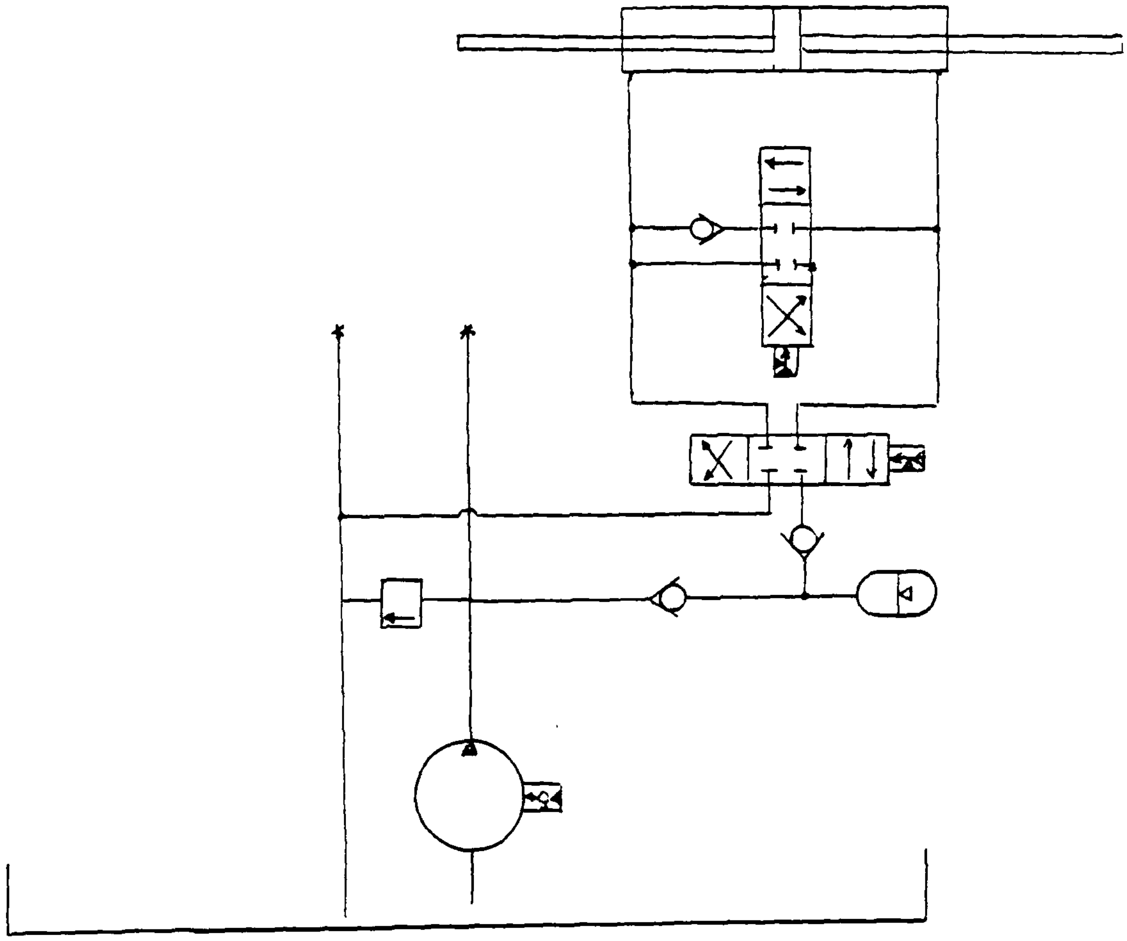

FIGURE 2. Shank Circuit Design, Original O.S.U. Leg Configuration

Preliminary Analysis

A preliminary analysis of the original O.S.U. circuit design was developed to evaluate it with regard to the system requirements. First, the relationship between cylinder size and system pressure was investigated using the definition of pressure, $P = \frac{F}{A}$, and a maximum load of 1800 lbs (see Table 1). In these calculations, area represents the net area of the cylinder over which the pressure is acting. To further illustrate this relationship, it was assumed that the system pressure acted on the entire piston area with zero back pressure and the required system pressure was calculated based on standard diameters. This relationship between cylinder area and pressure is graphically illustrated in Figure 3.

This relationship can be further developed into a relationship between the pressure and the extension rate of the cylinder that can be used in selecting a system pressure. As a first approximation,

$$\frac{Q}{A} \times \frac{231\frac{in^3}{gal}}{60\ \frac{sec}{min}} = \dot{X}$$

where Q = flow (gal/min)

A = area (in^2)

$\dot{X}$ = cylinder extension rate (in/sec)

If the previous relation is substituted,

$$P = F/A$$

$$A = F/P = 1800\ lbs/P$$

$$\frac{Q \times P}{1800} \times \frac{231}{60} = \dot{X}$$

$$\frac{Q \times P}{467.53} = \dot{X}$$

Some representative values have been calculated and have been included in Table 2. The relationship has been graphically illustrated in Figure 4.

TABLE 1. Shank Circuit Preliminary Hydraulic Flow Calculations

Given: maximum load = 1800*

$$P = \frac{F}{A} \qquad A = \frac{F}{P} = \frac{1800}{P}$$

Pressure	Area	Diameter*
1000	1.8	
1500	1.0	
2000	0.9	
2500	0.72	
3000	0.6	
4074	.442	3/4 "
2292	.785	1"
1811	.994	1 1/8"
1467	1.227	1 1/4"
1212	1.485	1 3/8"
1018	1.767	1 1/2"

* Note: Values represent cylinder bore diameters with zero pressure on the rod end of the cylinder.

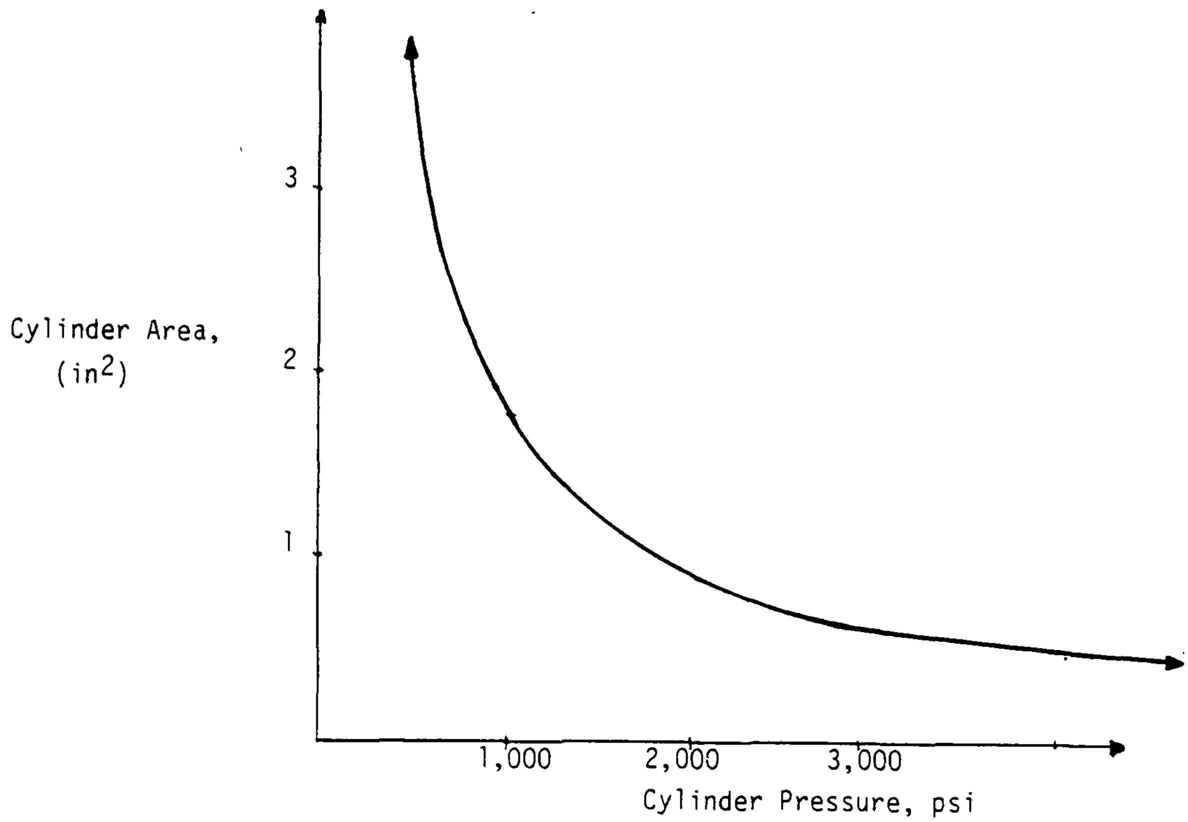

FIGURE 3. Cylinder Area vs. Pressure Relationship for 1800 Pound Load

TABLE 2. Cylinder Extension Rate
as a Function of Pressure and Flow

Cylinder Pressure, psi	Extension Rate (in/sec)				
	5	10	15	20	30
1,000	10.69	21.39	32.08	42.78	64.17
1,500	16.04	32.08	48.13	64.17	96.25
2,000	21.39	42.78	64.17	85.56	128.33
2,500	26.74	53.47	80.21	106.94	160.42
3,000	32.08	64.17	96.25	128.33	192.50

At this point, some basic assumptions regarding the system were made. First, it was assumed that the shank circuit had 15 g.p.m. of flow available to it. Second, it was assumed that, in cruise mode, three legs would be operated simultaneously, therefore each cylinder would have 5 g.p.m. available to it. Third, flow rates could be increased up to a factor of 2 by incorporating accumulators into the system. Based on these assumptions, each leg would be provided between 5 and 10 g.p.m. to extend 18 inches in 0.05-0.10 seconds ($\dot{X}$ = 180 - 360 in/sec). As can be seen in Figure 4, these operational requirements are not within capabilities of the system under consideration.

After further discussions with Prof. Waldron at O.S.U., the system requirements were revised to the following:

<u>Cruise Mode</u>

1) Retraction - lift foot a maximum of 12 inches in 0.25 seconds
2) Extension - must be powered down to extend the foot 12 inches in approximately 0.25 seconds
3) Lock on - no extension of the foot under load for approximately 0.5 seconds.

All other requirements remained unchanged.

With 15 gpm available to the circuit and in light of these revised requirements, the following conclusions can be drawn from Figure 4:

1) a system pressure of 1000 psi is completely inadequate to meet the revised system requirements under the original leg configuration
2) a system pressure of 2,000 psi, with accumulators, might be marginally adequate to meet system requirements, within the accuracy of these assumptions
3) a system pressure of 3,000 psi with accumulators, would be capable of meeting system requirements under the original leg configuration.

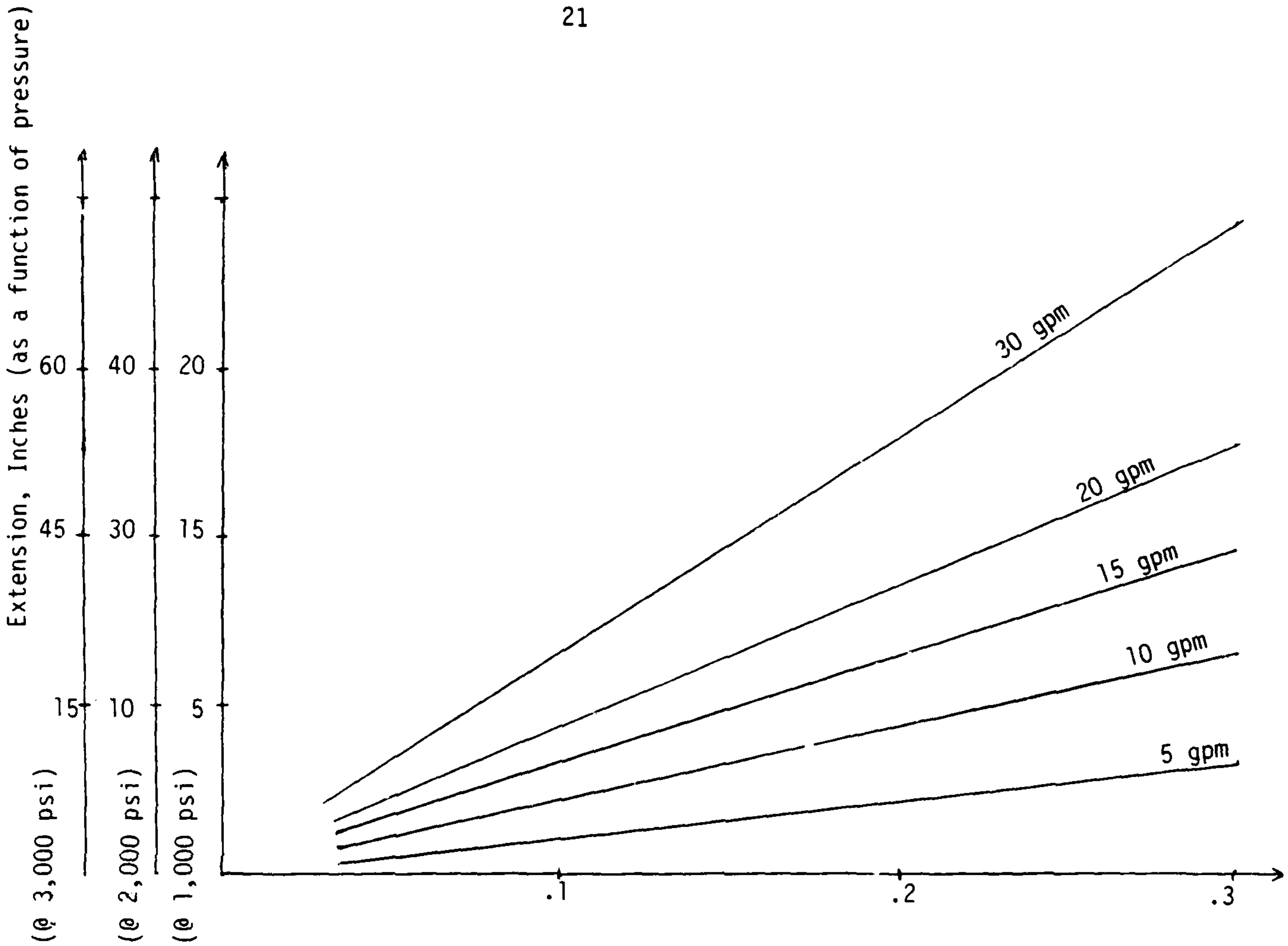

FIGURE 4. Cylinder Extension vs. Time

As a result of this analysis, it was agreed with Prof. Waldron at O.S.U. that the system pressure for the shank circuit should be 3,000 psi.

A second approximation analysis was then developed to further evaluate this circuit design based on the revised system requirements. This second approximation incorporated the inertial effects of the leg and the pressures losses due to the valve, hoses, and fittings into the system model. An iterative model was developed to calculate the maximum flow rate possible under given conditions. First, according to Newton's law,

$$F = M \frac{d^2x}{dt^2}$$

where F = force on piston rod
M = mass of lower leg
x = cylinder extension (ft)
t = time (sec)

$$\text{Area} \times (P_{pump} - \Delta P_{losses}) = M \frac{d^2x}{dt^2}$$

From the definition of the C_v factor

$$Q' = C_v \sqrt{\frac{\Delta P}{S.G.}}$$

where Q' = flow (in^3/sec)
S. G. = specific gravity of fluid

$$\Delta P_{losses} = \frac{S.G. \times Q'^2}{C_v}$$

$$P_{pump} - \frac{S.G. \times Q'^2}{C_v{}^2} = \frac{M}{\text{Area}} \quad \frac{d^2x}{d^2t}$$

$$\text{Also } Q\ (in^3/sec) = \text{Area} \times \frac{dx}{dt}$$

Therefore,
$$P_{pump} - \frac{S.G. \times \text{Area}^2 \times \left(\frac{dx}{dt}^2\right)}{C_v{}^2} = \frac{M}{\text{Area}} \quad \frac{d^2x}{dt^2}$$

$$\frac{d^2x}{dt^2} = a = \frac{A}{M} \quad P_{pump} - \left[\frac{S.G. \times Area^2 \times \left(\frac{dx}{dt}\right)^2}{C_v^2}\right]$$

This equation can be solved iteratively for acceleration by substituting into the equation a value for the velocity $\frac{dx}{dt}$ that is either an assumed initial value or previously calculated. Once the acceleration for a given time step has been calculated in this fashion, the velocity $\frac{dx}{dt}$ and displacement (x) for that time step can be calculated:

$$v = v + at$$

$$x = x + vt$$

This iteration is carried out until the values of acceleration approach zero, indicating that the system has reached steady state.

An assumed system was analyzed consisting of the following hydraulic components:

1) Moog 15 gpm servo valve (A076-104)
2) Schroeder NF30 filter with N10 element
3) Forty feet of 1/2 in. schedule 80 pipe (convient and conservative model for 1/2 in. tubing)
4) Four 1/2" elbows
5) Ten feet of 1/2" hydraulic hose

It was conservatively assumed that 15 gpm flowed through all components. A C_v factor of 1.6 was calculated for the overall system (see Appendix A).

In these equations, a pump pressure was assumed and a piston area was calculated based on the pressure. Initial values of zero were assumed for both the velocity and displacement. Some preliminary calculations were carried out with a hand calculator for pump pressures of 1500, 2500, and 3200 psi. The results of these calculations indicate that the system will reach steady state rapidly and, therefore, the inertial effects of the leg will not significantly impact the design of the hydraulic system for this leg. This analysis is currently being programmed and detailed results will be incorporated into future reports.

Finally, component manufacturers were contacted to determine representative response times for the servo valve and the accumulators.

According to the manufacturer, the 15 gpm Moog servo valve will either fully open or fully shut in 50 milliseconds. A 5 gpm valve is also available that requires even less time. It is anticipated that a reasonably fast operating time (such as 50 milliseconds) can be compensated for to a certain degree by computer control of the valve actuation.

Greer Hydraulics was also contacted for information regarding hydraulic accumulators. No information was readily available concerning the time required to discharge. Some limited testing may be required to obtain this information. Information was received on both the adiabatic and isothermal relationships between pressure and volume for a one gallon Greer accumulator that will be useful in modeling the hydraulic circuit.

Following the preliminary analysis of the shank circuit for the original O.S.U. leg design, Battelle was informed by O.S.U. that major revisions of the leg design were probable. Therefore, further analytical efforts were postponed pending this decision.

Demonstration System

Battelle was supplied by O.S.U. with a 15 gpm Hydura pump, with its associated servo control and electronic controller. This system was assembled, with a hydraulic reservoir and hydraulic motor, into a preliminary shank circuit demonstration unit. The purpose of this demonstration was to verify the performance of the pump and controller and to illustrate the basic control concept. This system was demonstrated to Dr. Clinton Kelly from DARPA at the program review in May. At this time, the pump appeared to operate as expected with one exception; the make-up pump (a Vicking gear pump) generated significantly more heat than was expected. It appeared to be attributable to a slight misalignment of the pump rotor with the housing. In effect, the rotor rubbing against the housing generating heat. Since the system filters had not yet been received from O.S.U., further investigation of this situation was postponed.

In June, high pressure filters and a Moog servo valve were incorporated into the demonstration system for the original O.S.U. design, as shown in the following hydraulic circuit drawing (see Figure 5). The system was then operated with the Hydura pump stroked in only one direction so that flow was always supplied to the pressure port of the Moog servo valve. The servo was driven by a DC power supply at voltages varying between zero and 3.2 volts, with the servo valve coils hooked up in parallel.

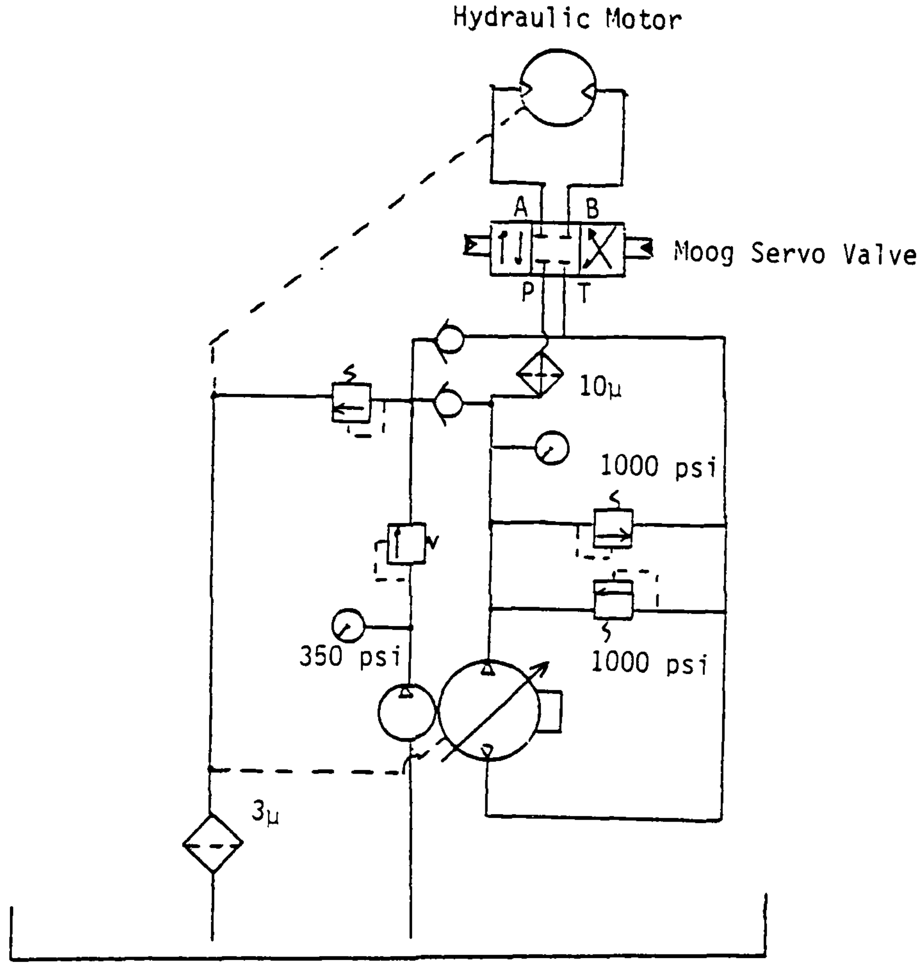

FIGURE 5. Hydraulic Circuit of Demonstration Unit

As power was applied to the Moog servo, the hydraulic motor would start turning and increase in speed with increasing voltage. By reversing the voltage to the servo coils, the motor could be run in the opposite direction. The Moog servo valve performed well in these tests.

The gear pump on the Hydura pump continued to run warm as before. Unlike before, it took a while to warm up (about 10 minutes for 140°F) which was much slower than before. This suggests that the pump is probably still "milling" its pressure plate but not as much as before. The return line filter showed considerable back pressure, indicating that a lot of dirt has been collected (possibly milling chips). Since the pump will be dismantled in the near future for shipment back to O.S.U., physical examination of the interior of the pump is being postponed until that time.

At the same time, a demonstration model of a regenerative hydraulic circuit was assembled. A regenerative circuit is one in which the rod end of the cylinder is connected to the blind end in such a way that allows rapid extension of the piston. This may have some application to the shank circuit, particularly in the case of an emergency extension.

Pump Review

A Vickers PVB 10 hydraulic pump was tentatively selected by the University of Wisconsin for the initial testing of the shank circuit. Specifications for this pump have been included in Appendix B. The specifications were reviewed and information on other, comparable pumps was accumulated. These pump characteristics are shown in Table 3. After reviewing the relative performance specifications for these pumps, it was concluded that while pumps were available with significant improvements in performance over the Vickers, these pumps had substantially higher costs and/or weights relative to the Vickers. Since the Vickers appeared to meet the preliminary requirements for the shank circuit, and since more detailed analyses were not possible until the selection of the final leg configuration, it was concluded that from the standpoint of the hydraulic design of the shank circuit, the Vickers pump was a reasonable unit to purchase.

While reviewing the Vickers pump, it became apparent that a supercharge pressure would be necessary at the suction inlet of the pump. A simple design for a supercharge circuit, shown in Figure 6, was developed. The supercharge circuit involves joining the return oil from the shank circuit and the return oil from the charge pumps for the hydrostatic transmission. By joining the circuits above a check valve, supercharge pressure can be supplied to the shank circuit pump. However, care must be taken to make sure the cooler is large enough so that supercharge oil does not exceed maximum allowed for pump.

TABLE 3. Pump Characteristics

Pump Model	Pressure (psi)	Flow (gpm)	Speed (rpm)	Price	Delivery	Weight
Sunstrand 22-2059 Scott Equipment (Mr. Trent)	320	53	3200	$2346	7-10 wks.	150 lbs
Denison PAV07-01C-041-3R-01-1A04 Hyd. & Air Controls (Ruth)	5000	44	2500	3570	1 week	110 lbs
Vickers AVB-10-FR-SY-CC (Pabco)	3000	18	3200	575	2 wks.	31 lbs
Rexroth A7V-28DR-2.0-RPF000 (Hydrotech - Harold Fraser)	5070	22	3000	1343	16 wks.	79 lbs
Rivett PV 2024-2659	2000	24	2400			56 lbs
Denison P46V-01-D-103-1R-03-1-04	5000	53	2500			

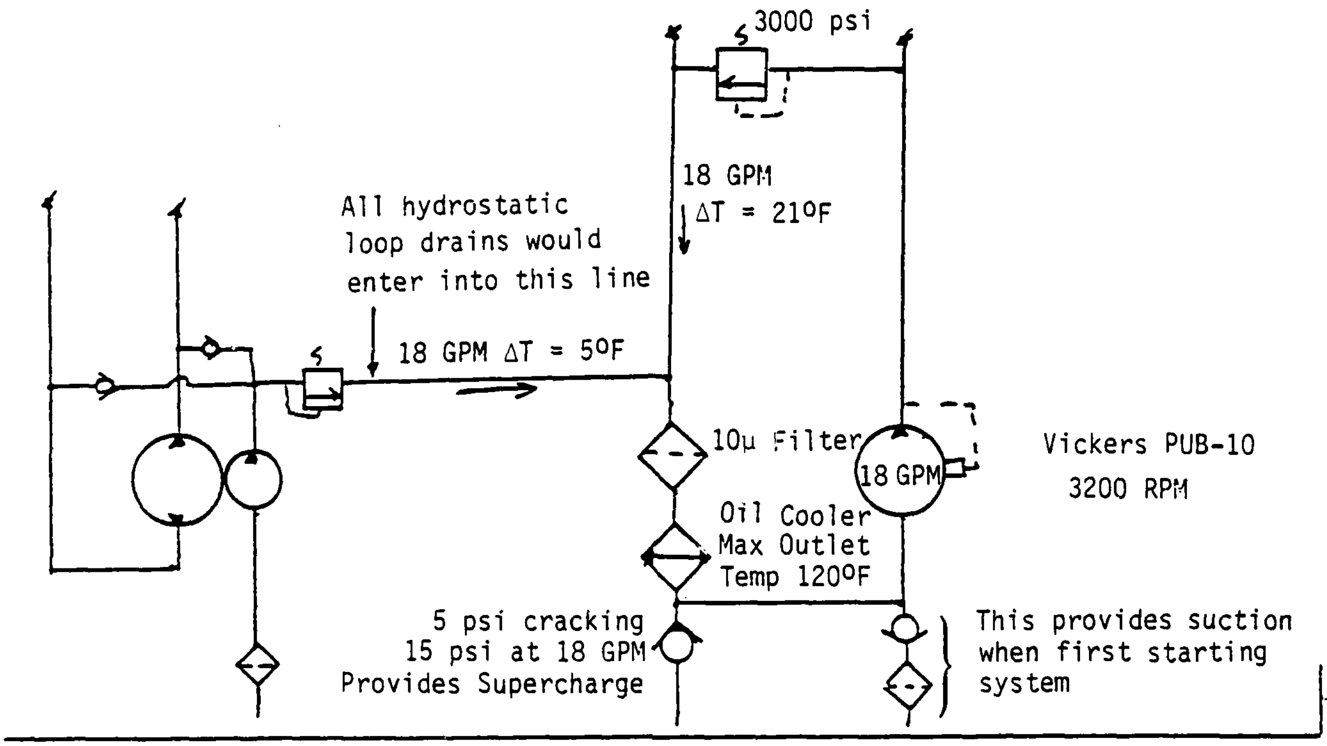

FIGURE 6. Supercharge Circuit for Vicker PVB 10 Pump

CONCLUSIONS

In the guidance portion of the Battelle project, work has been performed during the first half of the project whose goal was to derive stepping algorithms to enable a legged vehicle to traverse rough terrain. An approach for that problem has been derived, and is currently being investigated in detail. During the remaining six months of the project, this work will involve the further development and improvement of the selected approach and the investigation of alternative methods.

The actuation portion of the project has been directed at translating vehicle system requirements into the preliminary design of an hydraulic shank circuit capable of performing the required maneuvers. Much of the preliminary work is complete, and the requirements generated by other DARPA contractors on the adaptively controlled vehicle program are being consolidated and translated into a system design. During the remainder of this program, this design will be explored in more depth and a series of demonstrations undertaken.

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4. McGhee, R. B., and Iswandhi, G. I. "Adaptive Locomotion of a Multi-legged Robot Over Rough Terrain". IEEE Transactions on Systems, Man, and Cybernetics, Vol. SMC-9, No. 4. April, 1979.

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APPENDIX A

CALCULATION OF SYSTEM C_V

APPENDIX A
CALCULATION OF SYSTEM C_V

Servo Valve

For the 15 gpm Moog servo valve (A076-104), vendor literature indicates a pressure drop of 1000 psi across the valve at a no-load flow of 15 gpm (see Figure A-1). Therefore:

$$15 \text{ gpm} \times \frac{236 \frac{in^3}{gas}}{60 \frac{sec}{min}} = C_V \sqrt{\frac{\Delta P}{S.G.}}$$

s.g. for hydraulic fluid ≃ 0.865

$$15 \times \frac{236}{60} \sqrt{\frac{0.865}{1000}} = C_V = 1.74$$

For 1/2" Hose

According to "Fluid Power Designers Lightening Reference Handbook (Figure A-2), $\Delta P = 3.15$ per foot for flows of 15.3 gpm for 1/2 in. hose.

Therefore,

$$C_V{}^2 \quad 15.3 \text{ gpm} \times \frac{236 \text{ in}^3/\text{gal}}{60 \text{ sec/min}} \sqrt{\frac{S.G.}{\Delta P}}$$

$S.G. = .865$ $\qquad$ $\Delta P = 3.15/\text{ft}$

$$C_V = 31.54/1 \text{ ft}$$

Since $\Delta P = 31.5$ psi for 10 feet of hose,

$$C_V = 9.97 \text{ for 10 feet of hose}$$

Filter (e.g., Schroeder NF30 With N10 Element)

From Figure A-3, $\Delta P = 4.5$ psi @ 15 gpm
Therefore,

$$C_V = 15 \text{ gpm} \times \frac{236}{60}\sqrt{\frac{.87}{4.5 \text{ psi}}} = 25.94$$

FLUID POWER DESIGNER'S LIGHTING REFERENCE HANDBOOK

HYDRAULIC CHARACTERISTICS

Unless specified otherwise, all performance parameters are given for valve operation on Mobil DTE-24 fluid at 100°F (38°C).

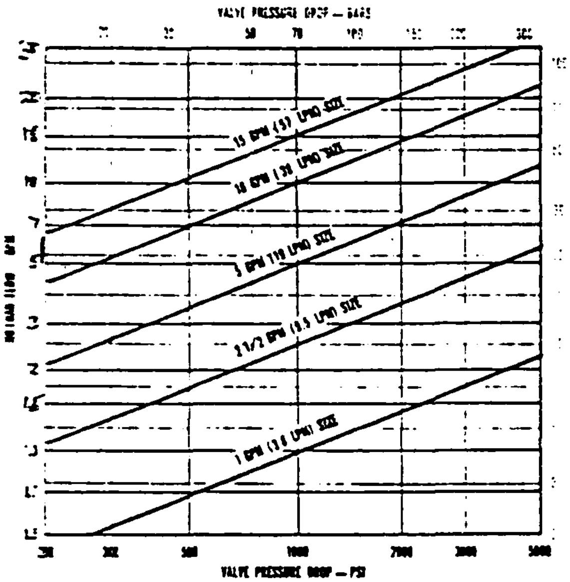


FIGURE 1 CHANGE IN RATED FLOW WITH PRESSURE

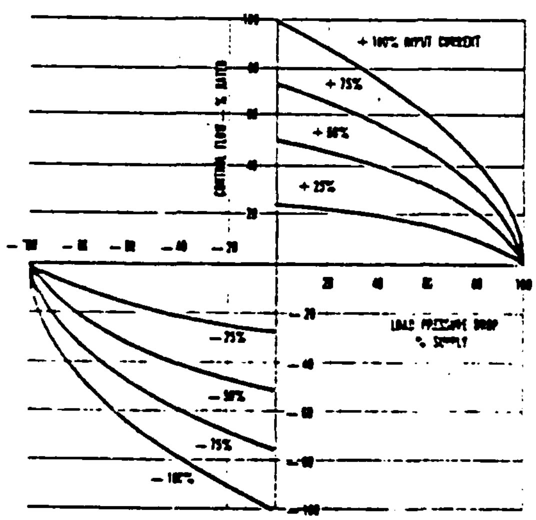


FIGURE 2 — CHANGE IN CONTROL FLOW WITH CURRENT AND LOAD PRESSURE

FLUID SUPPLY A076 Servovalves are intended to operate with constant supply pressure.

Supply Pressure	
minimum	200 psi (14 bars)
maximum standard	3000 psi (210 bars)
maximum special order	5000 psi (350 bars)
Proof Pressure	
at pressure port	150% supply
at return port	100% supply
NFPA static pressure rating* (test pressure 10,700 psi)	6900 psi
NFPA cyclic pressure rating (pressure port)* (cyclic test pressure 4350 psi for > 10^6 cycles)	3000 psi
Fluid†	petroleum base hydraulic fluids 60-450 SUS @ 100° F (10-97 cST @ 38° C)
Supply filtration required	10μm nominal (25μm absolute) or finer recommended
Operating temperature	
minimum	− 40° F (− 40° C)
maximum (unless limited by fluid)	+ 275° F (+ 135° C)

*Method of verifying static and fatigue pressure ratings per NFPA/T2.6.1-1974, category 3/90.

†Buna N seals are standard; Viton A and EPR available on special order.

RATED FLOW Five standard sizes are available having rated flows of 1, 2½, 5, 10, and 15 gpm at 1000 psi valve drop (3.8, 9.5, 19, 38, and 57 lit/min at 70 bars). See plot at left for corresponding rated flows at other supply pressures.

Flow with various combinations of supply pressure and load pressure drop can be determined by calculating the valve pressure drop.

$$P_V = (P_S - P_R) - P_L$$

P_V = valve pressure drop
P_S = supply pressure
P_R = return pressure
P_L = load pressure drop

FLOW-LOAD CHARACTERISTICS Control flow to the load will change with load pressure drop and electrical input as shown in Figure 2. These characteristics follow closely the theoretical square-root relationship for sharp-edged orifices, which is

$$Q_L = K\, i \sqrt{P_V}$$

Q_L = control flow
K = valve sizing constant
i = input current
P_V = valve pressure drop

INTERNAL LEAKAGE Maximum internal leakage for each size servovalve is:

Flow with 1000 psi (70 bars) Supply

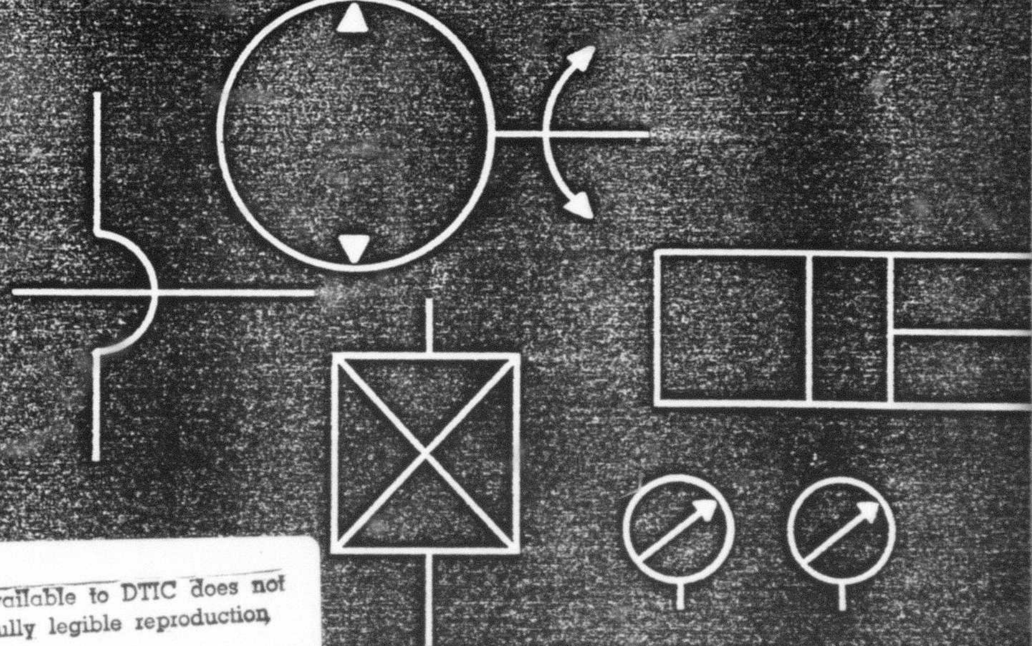
Rated Flow	Internal Leakage
1 gpm (3.8 lit/min)	< 0.17 gpm (0.66 lit/min)
2½ gpm (9.5 lit/min)	< 0.22 gpm (0.83 lit/min)
5 gpm (19 lit/min)	< 0.35 gpm (1.32 lit/min)
10 gpm (38 lit/min)	< 0.35 gpm (1.32 lit/min)
15 gpm (57 lit/min)	< 0.35 gpm (1.32 lit/min)

FIGURE A-1. Servo Valve Data Sheet

MATERIAL						PRESSURE LOSS (PSI/FOOT LENGTH) IN PIPES AT AVERAGE FLOW VELOCITY (FT/SEC) OF														EQUIVALENT PIPE LENGTHS (FT.) FOR CIRCUIT COMPONENTS					
SIZE (INCHES)	PIPE / HOSE	O.D. INCHES	I.D. INCHES	WALL INCHES	I.D. AREA SQ. IN.	5		7		10		15		20		25		30		TEE			ELBOW		
						LOSS	GPM	LOSS	GPM	LOSS	GPM	LOSS	GPM	LOSS	GPM	LOSS	GPM	LOSS	GPM						
1/8	PIPE-SCH 40	.405	.269	.068	.057	1.25	.89	1.79	1.24	2.60	1.75	3.16	2.67	5.47	3.56	6.20	4.45	7.07	5.34						
	PIPE-SCH 80	.405	.215	.095	.036	1.89	.56	3.05	.78	4.26	1.12	5.20	1.68	8.38	2.24	11.1	2.80	12.7	3.36						
	HOSE	—	.125	—	.012	5.96	.186	8.37	.260	11.9	.372	18.0	.558	24.0	.744	30.0	.930	35.7	1.11						
1/4	PIPE-SCH 40	.540	.364	.088	.104	.67	1.62	1.05	2.27	1.64	3.24	1.92	4.96	2.97	6.48	3.23	8.10	3.73	9.72						
	PIPE-SCH 80	.540	.302	.119	.072	1.11	1.12	1.49	1.57	2.11	2.24	2.84	3.36	4.15	4.48	5.08	5.60	6.30	6.72						
	HOSE	—	.250	—	.049	1.57	.758	2.17	1.08	3.00	1.49	4.12	2.23	6.04	2.98	7.49	3.72	8.95	4.44						
3/8	PIPE-SCH 40	.675	.493	.091	.191	.39	2.98	.57	4.18	.86	5.96	1.05	8.94	1.69	11.92	4.27	14.9	5.78	16.9	2.7	.8	2.7	1.2	2.7	.6
	PIPE-SCH 80	.675	.423	.126	.140	.54	2.18	.74	3.06	1.10	4.36	1.34	6.54	1.97	8.72	5.19	10.9	7.20	13.1						
	HOSE	—	.375	—	.110	.685	1.71	.97	2.43	1.34	3.35	2.02	5.03	2.88	6.71	3.33	8.38	3.99	10.0						
1/2	PIPE-SCH 40	.840	.622	.109	.304	.24	4.74	.36	6.65	.49	9.48	.68	14.22	2.09	18.98	3.38	23.7	4.28	28.4	3.5	1.05	3.5	1.5	3.5	.75
	PIPE-SCH 80	.840	.546	.147	.234	.30	3.65	.45	5.12	.71	7.30	.78	10.9	2.47	14.6	3.61	18.2	5.00	21.9	2.9	.9	2.9	1.4	2.9	.68
	PIPE-SCH XX	.840	.252	.294	.050	1.54	.78	2.19	1.09	3.[illegible]	1.56	3.65	2.34	6.13	3.12	7.48	3.90	9.55	4.68						
	HOSE	—	.500	—	.196	.387	3.03	.547	4.30	[illegible]	5.94	1.13	8.90	2.4	11.9	3.15	15.3	4.5	17.7						
3/4	PIPE-SCH 40	1.050	.824	.113	.533	.14	8.32	.22	11.7	.27	16.6	.78	25.0	1.47	33.3	2.19	41.6	3.00	49.9	4.5	1.4	4.5	2.1	4.5	1.0
	PIPE-SCH 80	1.050	.742	.154	.432	.16	6.74	.26	9.45	.37	13.5	.87	20.2	1.71	27.0	2.48	33.7	3.52	40.4	4.0	1.2	4.0	1.6	4.0	.8
	PIPE-SCH XX	1.050	.434	.308	.148	.53	2.31	.67	3.24	1.05	4.62	1.31	6.93	1.94	9.24	5.06	11.6	7.02	13.9						
	HOSE	—	.750	—	.442	.171	6.82	.248	9.92	.336	13.4	.502	20.1	1.33	26.8	2.02	33.4	2.90	41.3						
1	PIPE-SCH 40	1.315	1.049	.133	.863	.10	13.5	.13	18.9	.34	26.9	.57	40.4	1.42	53.8	1.64	67.3	2.24	80.7	5.7	1.7	5.7	2.6	5.7	1.2
	PIPE-SCH 80	1.315	.957	.179	.719	.11	11.2	.15	15.7	.24	22.4	.62	33.6	1.23	44.8	1.84	56.1	2.93	67.3	5.2	1.6	5.2	2.5	5.2	1.1
	PIPE-SCH XX	1.315	.599	.358	.863	.26	4.39	.37	6.16	.53	8.78	.67	13.2	2.25	17.6	3.29	22.0	4.20	26.3	3.0	1.0	3.0	1.5	3.0	.75
	HOSE	—	1.00	—	.785	.097	12.2	.136	17.1	.194	24.4	.610	36.6	.987	48.8	1.51	61.2	2.02	73.4						
1-1/4	PIPE-SCH 40	1.660	1.380	.140	1.496	.05	23.4	.08	31.7	.25	46.7	.39	70.1	.78	93.4	1.18	117	1.47	140	7.5	2.4	7.5	3.7	7.5	1.6
	PIPE-SCH 80	1.660	1.278	.191	1.280	.07	20.0	.09	28.1	.26	39.9	.44	58.9	.85	79.8	1.27	99.8	1.80	120	7.0	2.1	7.0	3.5	7.0	1.5
	PIPE-SCH XX	1.660	.896	.382	.630	.13	9.83	.16	13.8	.24	19.7	.71	29.5	1.35	39.3	2.01	49.2	2.76	59.0	4.9	1.5	4.9	2.3	4.9	1.1
	HOSE	—	1.25	—	1.23	.062	19.1	.087	26.8	.125	38.2	.436	57.3	.738	76.4	1.08	95.5	1.52	115						
1-1/2	PIPE-SCH 40	1.900	1.610	.145	2.046	.04	31.8	.06	44.5	.19	63.5	.33	95.3	.64	127	.96	159	1.26	191	9.0	2.8	9.0	4.3	9.0	2.0
	PIPE-SCH 80	1.900	1.500	.200	1.767	.04	27.6	.06	38.6	.21	55.1	.42	82.7	.71	110	1.06	138	1.36	166	8.2	2.6	8.2	4.2	8.2	1.8
	PIPE-SCH XX	1.900	1.100	.400	.950	.09	14.8	.09	20.8	.32	29.6	.51	44.4	1.05	59.2	1.51	74.1	2.14	88.9	6.5	2.0	6.5	3.0	6.5	1.4
	HOSE	—	1.50	—	1.77	.044	27.7	.061	38.6	.180	55.1	.353	82.7	.59	110	.86	138	1.21	166						
2	PIPE-SCH 40	2.375	2.067	.154	3.355	.03	52.3	.08	73.4	.14	105	.24	159	.48	209	.69	262	.85	324	11.0	3.5	11.0	5.5	11.0	2.5
	PIPE-SCH 80	2.375	1.939	.218	2.953	.03	46.0	.09	64.6	.15	92.0	.26	138	.52	184	.73	230	.98	275	10.8	3.4	10.8	5.0	10.8	2.4
	PIPE-SCH XX	2.375	1.503	.436	1.773	.04	27.7	.12	38.8	.21	55.3	.36	82.9	.72	111	1.34	138	1.36	166	8.2	2.6	8.2	4.0	8.2	1.8
	HOSE	—	2.00	—	3.14	.024	48.9	.034	68.6	.123	97.8	.256	147	.41	196	.60	245	.80	293						
2-1/2	PIPE-SCH 40	2.875	2.469	.203	4.788	.03	74.8	.07	105	.11	149	.20	224	.37	299	.53	374	.72	449	14.0	4.2	14.0	6.5	14.0	3.0
	PIPE-SCH 80	2.875	2.323	.276	4.238	.03	66.1	.07	92.6	.12	132	.21	198	.39	264	.57	331	.87	397	13.0	4.0	13.0	6.1	13.0	2.9
	PIPE-SCH XX	2.875	1.771	.552	2.464	.03	38.5	.10	53.4	.17	76.9	.30	115	.59	154	.79	193	1.15	231	10.3	3.1	10.3	4.8	10.3	2.2
	HOSE	[illegible]	2.50	[illegible]	4.91	.016	76.5	.045	107	.09	153	.18	229	[illegible]	306	[illegible]	382	.617	459						

Fluid Power Designers' Lightning Reference Handbook

STANDARD ENGINEERING DATA



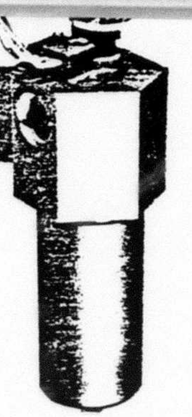
THIRD EDITION

FIRST IN FLUID HORSEPOWER

PAUL-MUNROE HYDRAULICS INC

Los Angeles (213) 723-0692   San Francisco (415) 697-2950   San Diego (714) 232-7600   Bakersfield (805) 832-3630

NF30 15 GPM/3000 PSI



3 to 60 MICRONS

Pressure Rating: 3000 PSI (204 Bar) operating
10,000 PSI (680 Bar) minimum yield

Maximum Operating Temperatures:
250°F (121°c) with N elements
300°F (149°c) with NM elements

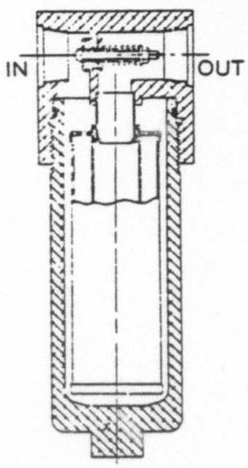
Bypass Valve Setting: 30 PSI (cracking pressure)

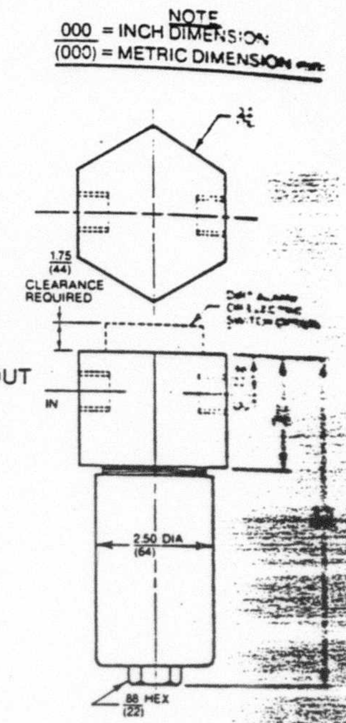
Material: Porting Head: Aluminum
Element Case: Aluminum

Compatibility: All commonly used mineral base and phosphate esters fluids. DF30 recommended for high water base and water glycol fluids. H seal adder recommended for phosphate esters. Consult factory for indicator availability for phosphate esters.

Servicing: Element case threaded into porting head. Minimum clearance required 2½" (64 mm.)

Weight: 3 lbs. (1 kg.)



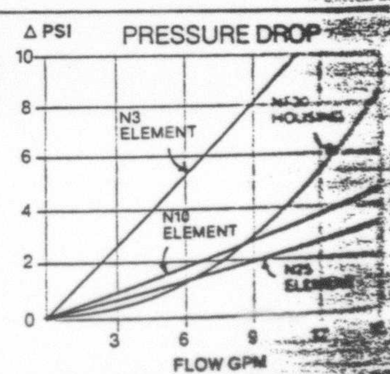


ORDERING INFORMATION

MODEL NUMBER SELECTION*

FILTER SERIES	NUMBER OF ELEMENTS	ELEMENT TYPE	SEAL MATERIAL	PORTING OPTIONS	DIRT ALARM TYPE
NF30	-1 = one element	N3 N10 N25 NM10 NM60	(Omit) = Buna N V = Viton	-P = ¾" NPTF -S = 1 1/16" - 12 SAE straight	(Omit) = none -D = color coded visual dirt alarm -MS2 = electrical switch

*Example: **NF30-1N10V-P-D**
NF30 filter with single 10 micron throw away element, viton seals, ¾" pipe ports, and visual indicator.



Element ΔP = (ΔP from curve) X (SSU Viscosity/150)
Filter ΔP = Housing ΔP + element ΔP
Element ΔP based on 150 SSU

SELECTION CHART

PRESSURE	ELEMENT Type	ELEMENT Micron	ENCLOSURES AND ELEMENTS TO MEET FLOW REQUIREMENTS (Based on use of 150 SSU mineral base hydraulic fluid)	
TO 3000 PSI	N Throw-Away	3	NF30-1N3-P	SEE MODEL DF30
		10	NF30-1N10-P	
		25	NF30-1N25-P	
	NM Clean-able	10	NF30-1NM10-P	
		60	NF30-1NM60-P	
Flow GPM			5	10

For 1/2" Pipe

For 1/2" pipe (schedule 80) (roughly the same I.D. as 1/2" tubing), using Figure A-2 again,

$$\Delta P = 2.47 \text{ psi @ } 14.6 \text{ gpm}$$

Therefore,

$$C_v = 14.6 \text{ gpm} \times \frac{236}{60} \sqrt{\frac{S.G.}{\Delta P}}$$

$$33.98 \text{ (for 1 ft)}$$

For 40 ft, $C_v = 5.37$

Elbows

For 4 1/2" elbows (schedule 80), each elbow is equivalent to 2.9 feet of 1/2" pipe (schedule 80). From Figure A-2, $\Delta P = 2.47$ psi/ft at 14.6 gpm.

$$C_v = 14.6 \times \frac{236}{60} \sqrt{\frac{.87}{4 \times 2.9 \times 2.47}}$$

$$= 9.98$$

To calculate the system C_v

$$\frac{1}{C_v^2 \text{ total}} = \sum \frac{1}{C_{v_i}^2}$$

Component	C_{v_i}
Valve	1.74
Hose (10 ft)	31.54
Filter	25.94
Pipe (40 ft)	5.37
Elbows (4)	9.98

$$\frac{1}{C_v^2 \text{ total}} = \frac{1}{(1.74)^2} + \frac{1}{(31.54)^2} + \frac{1}{(25.94)^2} + \frac{1}{(5.37)^2} + \frac{1}{(9.98)^2}$$

C_v total = 1.6

APPENDIX B

TECHNICAL SPECIFICATIONS ON

VICKERS PVB 10 HYDRAULIC PUMP

APPENDIX B

TECHNICAL SPECIFICATIONS ON VICKERS PVB 10 HYDRAULIC PUMP

Pump Specifications

Vickers pumps PVB 10 with pressure compensator and maximum stroke adjustor (all mechanical settings with adjustment screws and nuts).

Pumps capable of 3200 rpm with 12 psig supercharge. Maximum pressure = 3000 psi; compensation range = 250 - 3000 psi/hput HP @ 1800 rpm and 3000 psi; extrapolated to 3200 rpm and 3000 psi, HP = 35.6, volume = 18 gpm, weight = 31 lb each.

SPERRY VICKERS TM

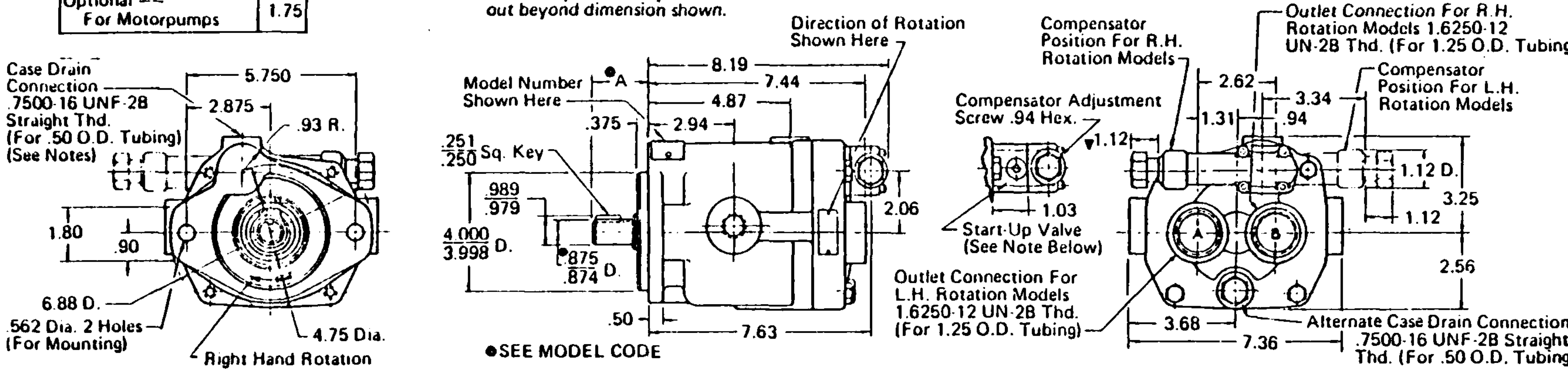
VARIABLE DISPLACEMENT PISTON PUMPS

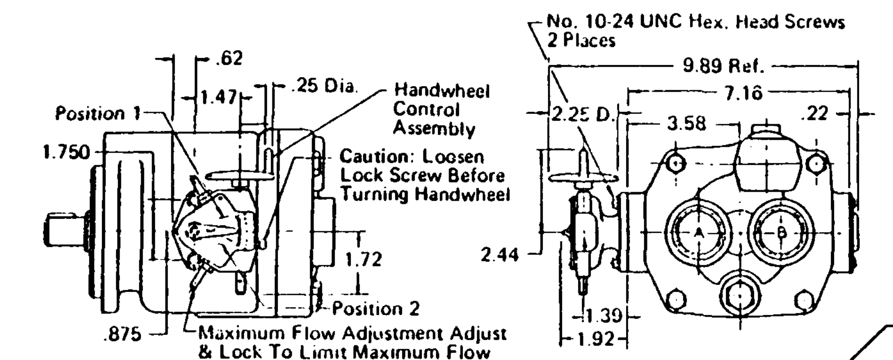
PVB10 AND PVB15 - 30 DESIGN (IN-LINE TYPE)
WITH PRESSURE COMPENSATOR, HANDWHEEL AND LEVER CONTROLS

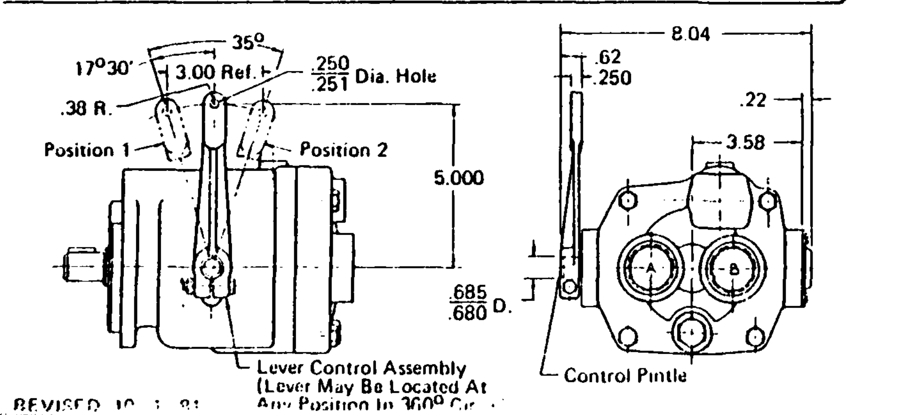
"CG" REMOTE COMPENSATOR SEE PAGE 508400B-1.

▼Caution – While pump is operating do not back compensator adjustment screw out beyond dimension shown.

Shaft Dimension "A" ●See Code For Models	
Standard	2.31
Optional — For Motorpumps	1.75







REVISED 10-1-81

HANDWHEEL CONTROL

Provides manual selection of pump displacement. Handwheel controlled units may be operated on both sides of center permitting bi-directional fluid flow characteristics.

Shaft Rotation	Pointer Position	Handwheel Rotation From Zero	Outlet Port
R.H.	1	Clockwise	A
R.H.	2	Counter-Clockwise	B
L.H.	1	Clockwise	B
L.H.	2	Counter-Clockwise	A

LEVER CONTROL

The lever control provides mechanical or manual selection of pump displacement. Lever controlled units may be operated on both sides of center permitting bi-directional fluid flow characteristics. Hand lever controls must be secured by suitable linkage arrangement to maintain desired setting. The control pintle may be rotated 17-1/2° on each side of center position to permit full reversal of flow. Pintle travel is limited to 35° by internal stops. Torque required to rotate control pintle is approximately 60 lbf·in at 1800 RPM and 1000 PSI. (Note: Torque varies with speed and pressure.)

Shaft Rotation	Lever Position	Outlet Port
R.H.	1	A
R.H.	2	B
L.H.	1	B
[illegible]	[illegible]	A

PRESSURE COMPENSATOR CONTROLS

The pressure compensator control automatically adjusts pump delivery to maintain volume requirements of the system at a preselected operating pressure. Maximum pump delivery is maintained to approximately 75 PSI (PVB10), 100 PSI (PVB15) below the pressure control setting before being reduced. The pressure compensator control operates on one side of center and has an adjustment range of 250 to 1500 PSI on "CM" models and 250 to 3000 PSI on "C" models as designated in the model coding. Customer must limit PVB15 to 2000 PSI when using "C" compensator.

WITH START-UP VALVE

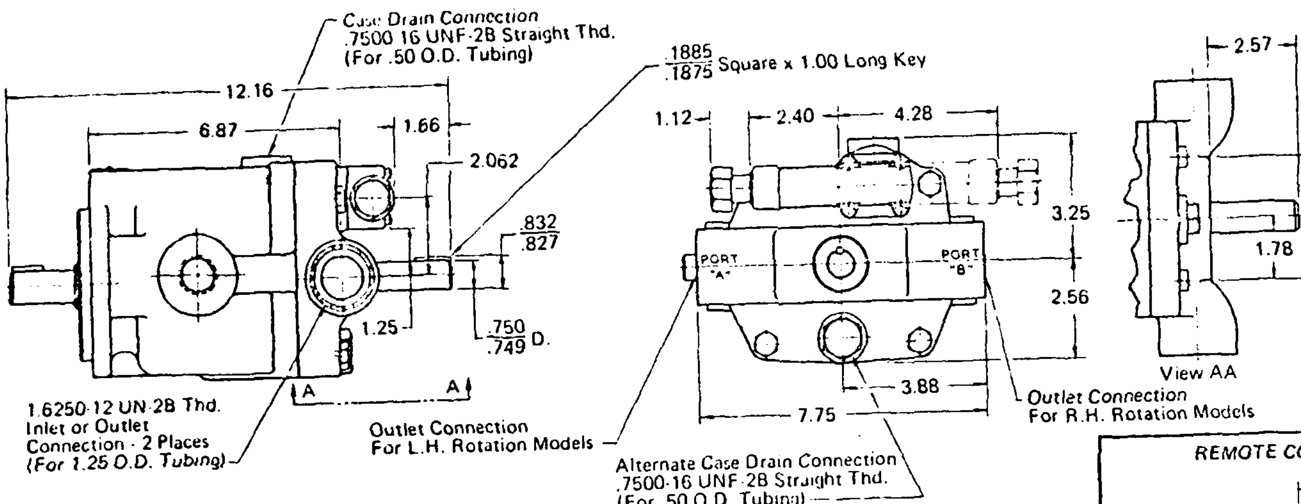
The pressure compensator control can be ordered with a start-up valve. This is a unique device that permits starting a pressure compensated pump with a smaller size electric motor than is normally required at rated conditions. It is designed for use with high flow-low pressure and low flow-high pressure applications in which torque requirements are considerably lower than at rated conditions. This device is completely automatic with no external signal needed.

Operational data for this device is as follows:

1. Start-up pressures can be doubled using a start-up valve and the same electric motor.
2. Electric motor size can be halved using a start-up valve with start-up pressures remaining the same.
3. Any application that does not meet the above statements must be referred to engineering.

SPERRY VICKERS VARIABLE DISPLACEMENT PISTON PUMPS

THRU SHAFT AND/OR SIDE PORTS
WITH PRESSURE COMPENSATOR, HANDWHEEL AND LEVER CONTROLS



PRESSURE COMPENSATOR CONTROL - THRU SHAFT AND SIDE PORTS

General Data

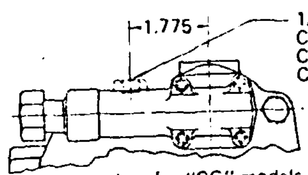
Side Ports

Rated and maximum speeds. Pressure and general performance of the side port pumps are identical to the standard units.

Thru Shaft

Rated and maximum speeds, pressure and general performance of the thru shaft pumps are identical to the standard units, except the input torque is limited to 735 lbf-in.

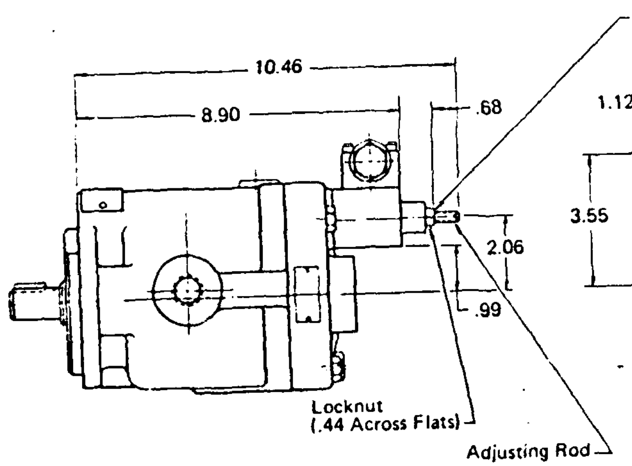
REMOTE COMPENSATOR "CG" (SEE MODEL CODE)

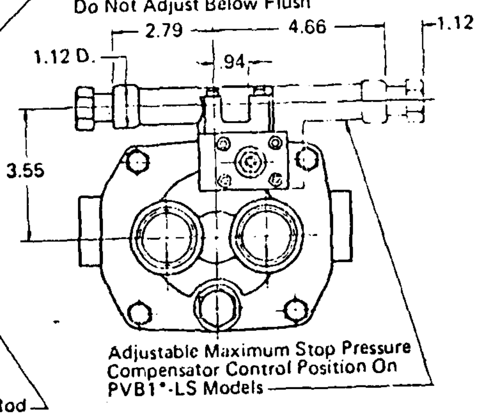


Adjustment procedure for "CG" models

1. Turn pressure control (such as C-175) CCW to minimum setting.
2. Turn compensator adjustment plug to desired minimum pressure (250 PSI or higher).
3. Full pressure range can now be obtained with pressure control.

Caution: Effective compensator pressure will be compensator control plus remote relief valve setting.





Control Description

The adjustable maximum stop pressure control enables the maximum pump delivery to be externally adjusted from 25% to 100% while maintaining all of the standard features of a pressure compensated pump. To assist initial priming, manual adjustment control setting must be at least 40 *percent of maximum flow position.*

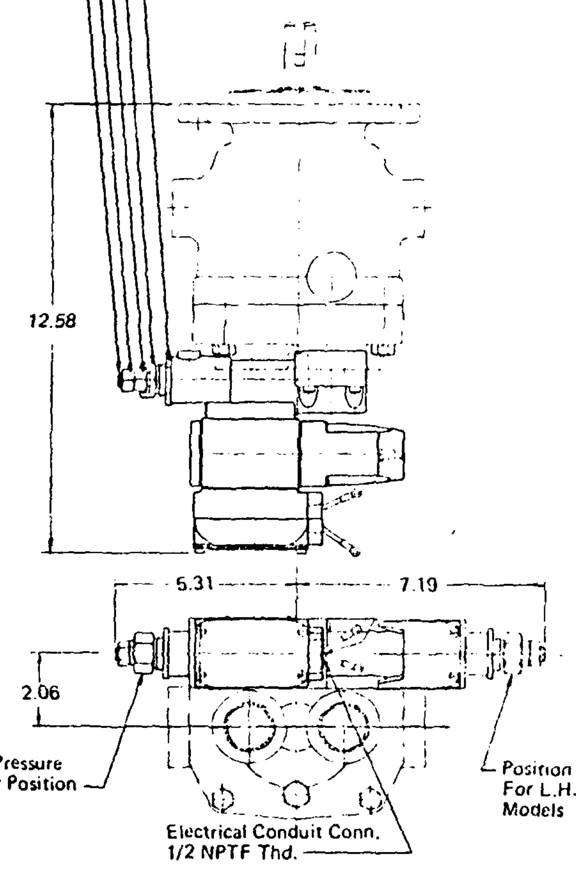
Control Adjustment

Loosen locknut on adjusting rod. Turn adjusting rod clockwise (CW) to decrease maximum pump delivery or counterclockwise (CCW) to increase maximum pump delivery until desired setting is obtained. Secure this setting by tightening locknut.

Note:
Not available with (X) thru shaft port models.

PRESSURE COMPENSATOR CONTROL WITH ADJUSTABLE MAXIMUM DISPLACEMENT STOP

5084008-1



ELECTRIC DUAL-RANGE
PRESSURE COMPENSATOR CONTROL

PVB10 AND PVB15 - 30 DESIGN VARIABLE DELIVERY INLINE PISTON PUMP WITH DUAL RANGE PRESSURE COMPENSATOR CONTROL

Control Description

The dual range pressure compensator control automatically adjusts pump delivery to maintain volume requirements of the system at either of two preselected operating pressures.

Maximum pump delivery is maintained to approximately 75 PSI (PVB10), 100 PSI (PVB15) below either of the pressure control settings before being reduced.

Electric control of the dual range pressure compensator is available in two pressure ranges of 250 to 1500 PSI or 250 to 3000 PSI. Control type and pressure range is designated in the model code.

Control Features

The dual range pressure compensator offers the following advantages:

1. Low - pressure setting for –
   A–*Low horsepower start-up*
   B–Tool or equipment tryout
   C–Low power consumption and low heat generation when the machine or circuit is at rest

2. High - pressure setting for –
   A–*Machining or circuit applications*

Control Adjustment

1. With the directional valve de-energized, loosen locknut "5" and turn the adjusting screw "4" to the desired first stage pressure setting and tighten locknut "5".

2. With solenoid de-energized, turn adjusting spool "1" counterclockwise (CCW) until nut "3" is *bottomed in adjusting screw slot*. (Second stage setting is now equal to first stage pressure setting). Turn adjusting spool clockwise (CW) to desired second stage pressure requirements. (One complete turn of adjusting spool equals approximately 600 PSI on "CD" models, and 300 PSI on "CMD" models.) Energize solenoid and check pressure setting. De-energize solenoid and re-adjust if necessary. Secure this setting by tightening locknut "2".

Note:
Not available with (X) thru shaft models.

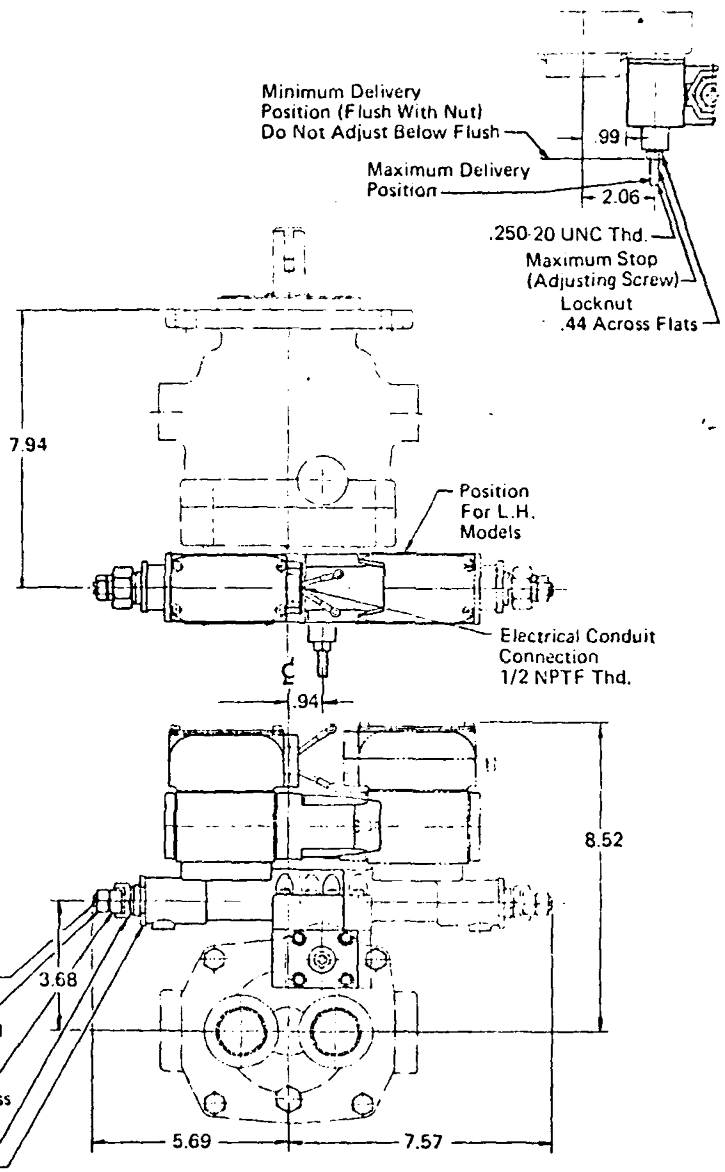
▲ Note:
Any sliding spool valve, if held shifted under pressure for long periods of time, may stick and not spring return due to fluid residue formation and, therefore, should be cycled periodically to prevent this from happening.

▲Solenoid data (110 V ac 50 Hz and 115/120 V ac 60 Hz)

Solenoid Current	Inrush amps (R.M.S.)✱	Holding amps
115/120 V ac 60 Hz – 110 V ac 50 Hz	2.0	.4

✱Maximum peak inrush amps approximately 1.4 x R.M.S. value shown.

508400B



ELECTRIC DUAL-RANGE PRESSURE COMPENSATOR AND MAXIMUM STOP CONTROL

PVB10 AND PVB15 - 30 DESIGN VARIABLE DELIVERY INLINE PISTON PUMP WITH DUAL RANGE PRESSURE COMPENSATOR CONTROL & MAXIMUM FLOW ADJUSTABLE STOP

Control Description

The dual range pressure compensator control with max. flow adjustable stop automatically adjusts pump delivery to maintain volume requirements of the system at either of two *preselected operating pressures.*

Maximum pump delivery is maintained to approximately 75 PSI (PVB10), 100 PSI (PVB15) below either of the pressure control settings before being reduced. The maximum delivery is adjustable and can be set at any setting from 25% to 100% flow.

Electric control of the dual range pressure compensator is available in two pressure ranges of 250 to 1500 PSI *or* 250 to 3000 PSI. Control type and pressure range is designated in the model code.

Compensator Control

1. With the directional valve *de-energized,* loosen locknut "5" and turn the adjusting screw "4" to the desired first stage pressure setting and tighten locknut "5".

2. With directional valve de-energized, turn adjusting spool "1" counter-clockwise until nut "3" is *bottomed in adjusting screw slot. (Second stage setting is now equal to first stage pressure setting).* Turn adjusting spool clockwise to desired second stage pressure requirements (one complete turn of adjusting spool equals approximately 600 PSI on "CCD" models, and 300 PSI on "CMCD" models). Energize solenoid and check pressure setting. De-energize solenoid and re-adjust pressure if necessary. Secure this setting by tightening locknut "2".

Maximum Flow Adjustment

With the system pressure below both compensator settings, *loosen* maximum stop adjusting screw locknut and adjust screw to desired flow position (turning screw clockwise decreases flow and turning screw counterclockwise *increases flow).* To lock screw in position tighten locknut. To assist initial priming, adjust control setting to at least 40% of maximum flow position.

Note:

Not available with (X) thru shaft models.

▲ Note:

Any sliding spool valve, if held shifted under pressure for long periods of time, may stick and not spring return due to fluid residue formation and, *therefore, should be cycled periodically* to prevent this from happening.

▲ Solenoid data (see note on page 508400B-2).

General Data

These units are of the axial piston, variable displacement, in-line design, displacement is varied by means of either a pressure compensator, handwheel, or lever control.

Installation Information

Horizontal mounting is recommended to maintain necessary case fluid level. The case drain line must be full size unrestricted and connected from the uppermost drain port directly to the reservoir in such a manner that the case remains filled with fluid. Piping of drain line must prevent siphoning. Pipe drain line so that it terminates below reservoir fluid level. No other lines are to be connected to this drain line. Caution must be exercised to never exceed 5 PSI unit case pressure.

Starting

Before starting, fill case with system fluid thru uppermost drain port. Case must be kept full at all times to provide internal lubrication.

When first starting, it may be necessary to bleed air from pump outlet line to permit priming and reduce noise. Bleed by loosening an outlet connection until solid stream of fluid appears. An air-bleed valve is available for this purpose. See drawing 521601.

To assist initial priming, control setting must be at least 40% of maximum flow position.

Operating Specifications

Model Number	Theoretical Displacement In.^{3}/Rev.	Delivery GPM At 1800 RPM	Delivery GPM At Max. RPM	Operating Speed RPM (Maximum)	Pressure PSI (Maximum)	■ Sound Data dB(A)	Input Horsepower At Max. PSI & 1800 RPM
PVB10	1.29	10	18	3200	3000	62	20
PVB15	2.01	15.7	26.2	3000	2000	65	21

Refer to curve for inlet pressure requirements.

■ Straight port model at cutoff, 1200 RPM, 2000 PSI pressure, SAE 10W oil at 120°F., 5" Hg inlet vacuum per NFPA standard T3.9.70.12.

Drive Rotation

Shaft rotation is not reversible and must be specified when ordering. (See drawing 517010 for indirect drive data).

Case Pressure

Not to exceed. 5 PSI

Handwheel Data

Approximate displacement for one turn of handwheel is .32 in.^{3}/rev. for 10 size and .5 in.^{3}/rev. for 15 size.

Filtration

Pressure or Return Line. . . . . 10 Micron Absolute or Less
Inlet. 149 Microns

Fluids

Clean petroleum antiwear industrial hydraulic oil, water-in-oil emulsions, HWBF, water glycol, phosphate esters, or automotive crankcase oil designated SC, SD or SE. Running viscosity range 70-250 SUS. Operating temeprature 120°F. recommended, 150°F. usual maximum. Refer to data sheet I-286-S for hydraulic fluid and temperature recommendations. Also see fluid life chart 507000.

This unit is designed to meet specifications as outlined. To insure maximum unit performance, in conjunction with your specific application, consult your application engineer if your:

Speed is above maximum RPM.

Fluid does not meet the specifications of data sheet I-286-S.

Mounting attitude is other than horizontal.

Needs require application assistance.

Model Code

F3 P V B 15 - (F) R S W Y - 30 -*C* (*) (L) - 10

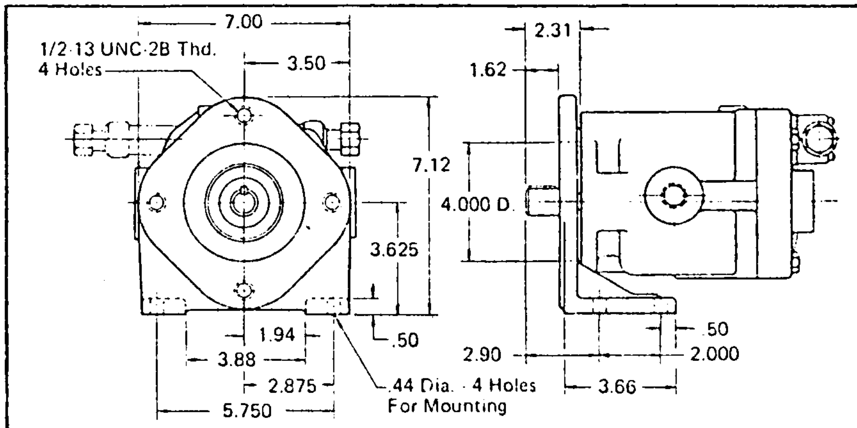
- Special Seals (Omit if not needed) — F3
- Pump — P
- Variable Displacement — V
- In-line Design — B
- GPM - Rating at 1800 RPM — 10 - 10 GPM   15 - 15 GPM
- Mounting — F - Foot Bracket (Omit for Flange). For Separate "Foot Bracket Kit" Order Model FB-B-10
- Rotation (Viewing Shaft End) — R - Right Hand; L - Left Hand
- Displacement — S - One Side of Center (Pressure Compensator Models); D - Both Sides of Center (Handwheel & Lever Models)
- Optional Ports & Shaft — W - Side Ports (Standard Drive Shaft); X - Side Ports and Thru Shaft (Standard Drive Shaft)
- ● Drive Shaft — Y - Standard (Omit For Short Shaft)
- Pump Design Number — Design Numbers Subject To Change. Installation Dimensions Remain As Shown For Design Numbers 30 Thru 39.
- *Control Type — C - Pressure Compensator (250-3000 PSI) See Compensator Control Information On Front Page. CM - Pressure Compensator (250-1500 PSI); H - Handwheel; M - Lever; V - No Control
- Control Option — C - Adjustable Maximum Displacement Stop (With Compensator) Not Available With Thru Shaft
- Compensator Variations — D - Dual-Range (Electric Control) (Not Available With Thru Shaft); E - Start-Up Valve; G - Remote Compensator (Use CCG For L.H. Side Port Models)
- Other Controls or Options — L - Left Hand Location Viewing Shaft End (H & M Only)
- Control Design Number — Design Numbers Subject To Change. Installation Dimensions Remain As Shown For Design Numbers 10 Thru 19, 20 Thru 29 For "D" Control

*Note: Pressure shown defines the minimum adjustable pressure range. It might be possible to make settings outside of this range.

Note: For thru-drive spline shafts, add S124 suffix to model code.

Weight Lb. (Approx.)

Flange Mounting. 31
Foot Mounting. 36



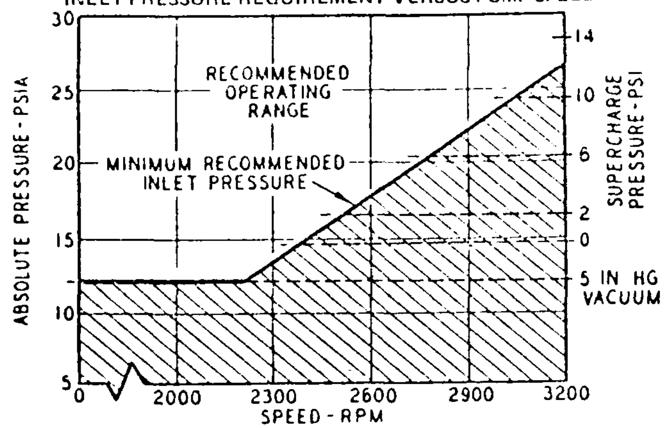
FOOT MOUNTING

508400B

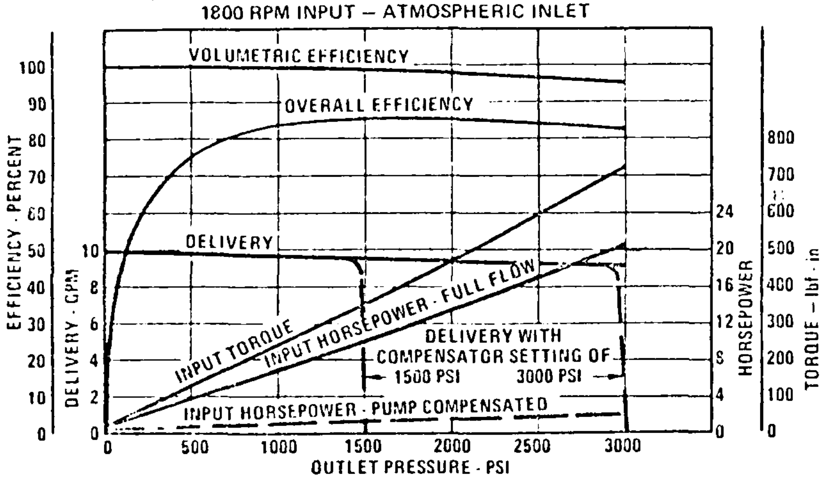
TYPICAL PERFORMANCE CURVES

PVB10 SERIES

INLET PRESSURE CURVE
BASED ON OIL TEMPERATURE OF 120° F (100 SUS)
VISCOSITY OF 150 SSU @ 100° F
INLET PRESSURE REQUIREMENT VERSUS PUMP SPEED

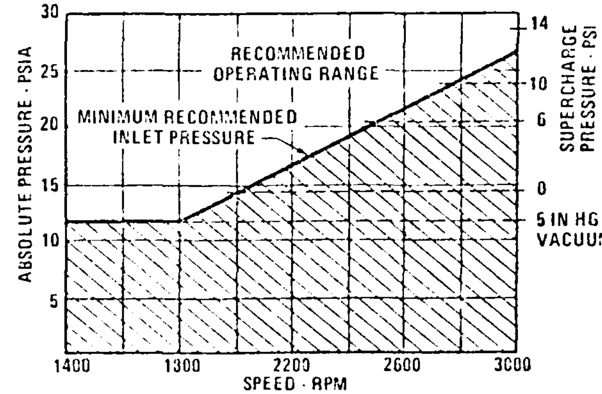


PERFORMANCE CHARACTERISTICS
BASED ON OIL TEMPERATURE OF 120° F (100 SUS)
1800 RPM INPUT — ATMOSPHERIC INLET

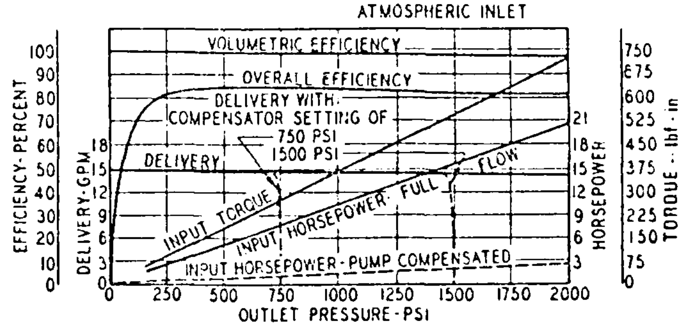


PVB15 SERIES

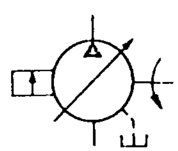
INLET PRESSURE CURVE
BASED ON OIL TEMPERATURE OF 120° F (100 SUS)
VISCOSITY OF 150 SSU @ 100° F
INLET PRESSURE REQUIREMENT VERSUS PUMP SPEED



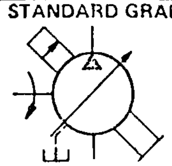
PERFORMANCE CHARACTERISTICS
BASED ON OIL TEMPERATURE OF 120° F (100 SUS)
1800 RPM INPUT — ATMOSPHERIC INLET



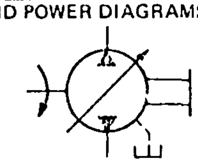
STANDARD GRAPHICAL SYMBOLS FOR FLUID POWER DIAGRAMS



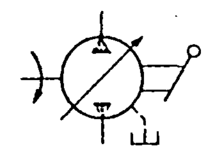
PRESSURE COMPENSATOR



PRESSURE COMPENSATOR
WITH MAXIMUM ADJUSTABLE STOP



HANDWHEEL CONTROL



LEVER CONTROL

50840